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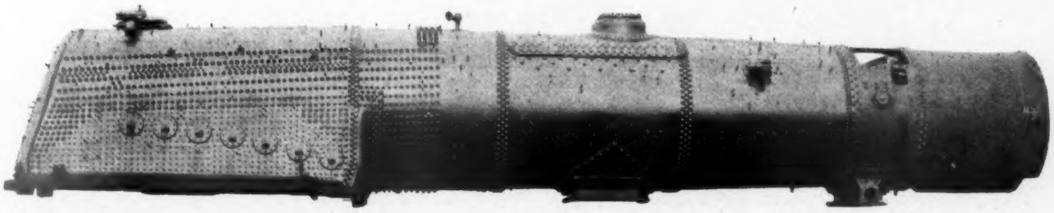

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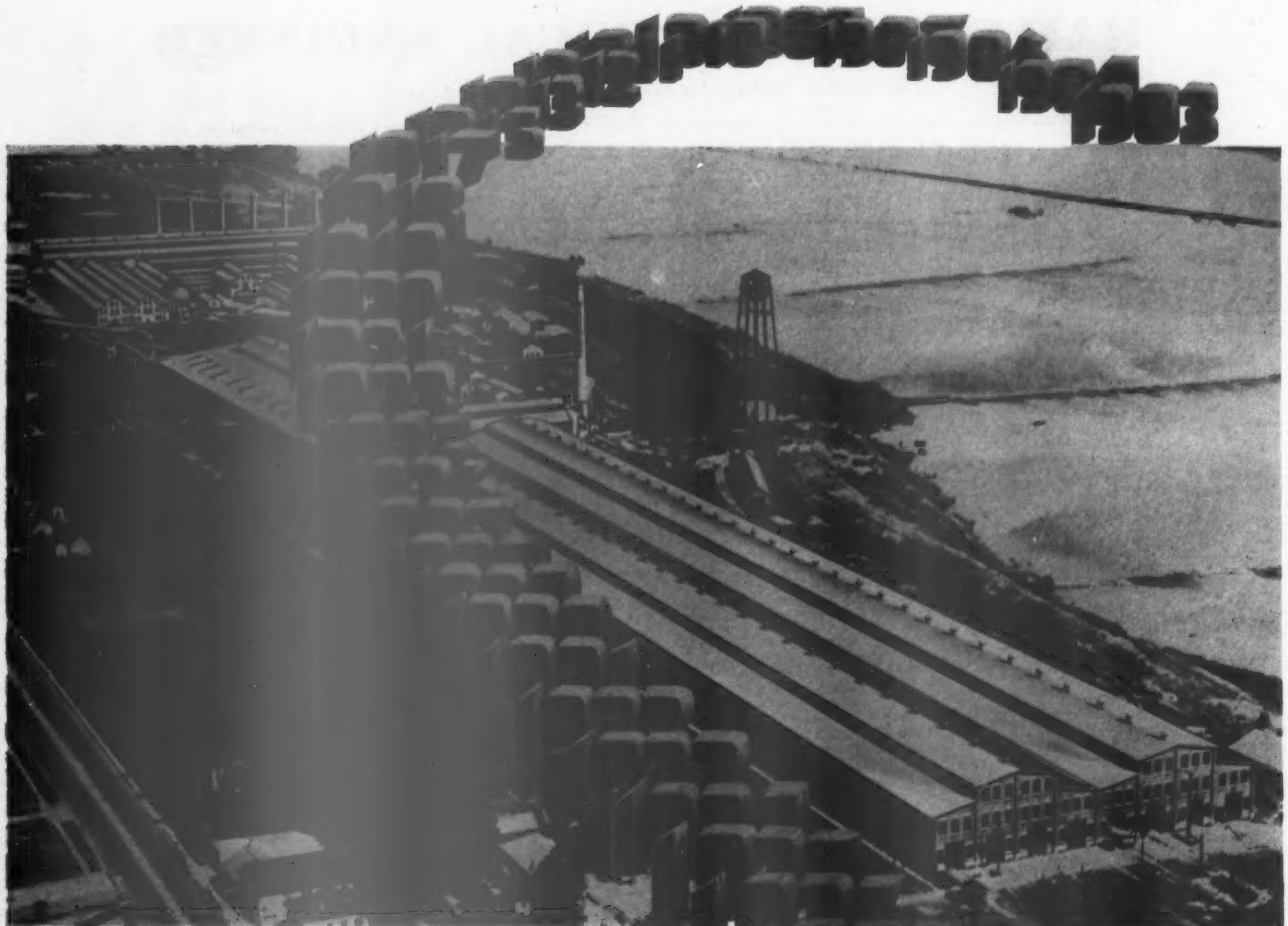

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RAILWAY
MECHANICAL ENGINEER

A Study of

The Locomotive Boiler*

By C. A. Brandt†

The author discusses some of the problems of boiler design and proportions that affect the efficiency and capacity of the conventional locomotive

The public, which is served by the railroads, is continuously demanding higher speeds for both freight and passenger trains, and this, in conjunction with a more intensive utilization of their motive power, is one of the many important problems confronting the managements of the American railroads today. Until recent years the steam locomotive had been the principal power unit, but its supremacy is now being challenged by other forms of motive power, particularly the electric- and Diesel-driven locomotives. To meet this challenge the designers of steam locomotives are constantly studying the problem of building boilers of greater steam-generating capacity within the permissible limits of size and providing engines of lowest posible steam consumption per unit of power.

The fact that higher boiler capacity for minimum weight has been an ever-present problem throughout the years of locomotive development in America can best be illustrated by citing the evolution of the eight-coupled locomotive. The first of this design, or the 2-8-0, utilized 90 per cent of the total locomotive weight for adhesion. The demands for higher speeds, requiring greater steam-making capacity, led to the addition of truck axles to carry the heavier boilers, which resulted in the successive development of the 2-8-2, the 4-8-4, and last the 6-4-4-6 type high-speed locomotive, exhibited at the New York World's Fair this year.

The total weight of an early-design 2-8-0 type locomotive with 270,000 lb., on the drivers and a tractive force of 67,500 lb. was only 300,000 lb. To supply the steam for the high-speed 6-8-6 type locomotive with the same weight on drivers and the same tractive force, requires a boiler of such size as to double the total weight of the locomotive to 600,000 lb., as shown in the table.

Increase in Weight of Eight-Coupled Locomotives Due to Larger Boiler Capacity

Type	Tractive force, lb.	Weight on drivers, lb.	Total weight of loco., lb.	Weight on drivers, per cent of total	Increase in weight of locomotive to gain additional boiler power: Actual, lb.	Per cent
2-8-0	67,500	270,000	300,000	90	......	..
2-8-2	67,500	270,000	360,000	75	60,000	20
4-8-2	67,500	270,000	400,000	67	100,000	33
4-8-4	67,500	270,000	450,000	60	150,000	50
6-4-4-6	67,500	270,000	600,000	45	300,000	100

As this and other modern locomotives are examples close to the maximum practical size that can be built, the important question is whether greater boiler efficiency is attainable, particularly at high-capacity operation, and what can be done to accomplish this.

The question of reducing the steam consumption of the engines is outside the scope of this paper, but the reduction of steam consumption per i. hp. from an average of 18 to 13 lb. is possible by the adoption of compound cylinders as reported from tests on French locomotives, as well as by the use of still higher superheat and valve gear adaptable for its utilization. This paper will be confined to a discussion of those problems of design believed to be most essential in the advancement of the art. To enhance the value of this contribution, Tables I to VI‡ are included which give the principal dimensions and boiler ratios of representative locomotives of various types built in America in recent years. Locomotive test curves are also presented.

The subject matter will be centered on problems affecting the design of the conventional locomotive fire-tube boiler only, not because of any belief that this type of boiler is the final answer to the locomotive steam generator, but because at present most of the locomotives in the world are equipped with this type of boiler. The best solutions to some of the problems encountered in the design of large boilers have not yet been agreed upon principally because of the lack of reliable test data. A committee of the Association of American Railroads submitted a report in 1939 recommending a conventional type of boiler for a proposed 6,400-hp. high-speed locomotive, which is an acknowledgment that the fundamental principles of this boiler design have proved practical and are the best available at the present time.

It is unnecessary to recite in detail the advantages of the conventional boiler; this has been done many times before. The virtues of the completely water-cooled radiant-heat-absorbing furnace, high gas velocities over the convection heating surfaces with forced draft, and high superheat, all originally inherent in the locomotive boiler, are now being recognized as essentials to efficient steam generation in other fields and are being rapidly adopted in stationary power-plant boilers.

Locomotive boilers of designs radically different from conventional construction have been built in the past, but so far none has proved sufficiently practical for general railroad service. There will soon be placed in service in America a different type of steam generator for operation in conjunction with condensing steam turbines, and still other types are under consideration.

* Paper presented on December 7, 1939, at the annual meeting of The American Society of Mechanical Engineers, at Philadelphia, Pa. This paper will be published in two parts, Part I appearing in this issue.

† Chief engineer, The Superheater Company.

‡ Tables I to III appear in this issue. Tables IV to VI will be published in a subsequent issue.

Table I—General Dimensions and Boiler Ratios of Six-Coupled Locomotives

1. Type	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4	4-6-4
2. Railroad	C.P.R.	C.B.&Q.	N.Y.C.	N.Y.C.	Can.Nat.	N.Y. N.H.&H.	B.&O.	A.T.&S.F.	C.M. St.P.&P.	C.&N.W.
3. Road class	H-1-C	S-4	J-1-B	J-3-A*	K-5-A	I-5	V-2	3460†	F-7	E-4
4. Date built	1937	1930	1927	1938	1930	1937	1935	1937	1938	1938
5. Boiler pressure, lb. per sq. in.	275	250	225	275	275	285	350	300	300	300
6. Cylinder diameter and stroke, in.	22x30	25x28	25x28	22½x29	23x28	22x30	19x28	23½x29	23½x30	25x29
7. Driving wheels, diameter outside tires, in.	75	78	79	79	80	80	84	84	84	84
8. Weight on drivers, lb.	186,800	207,730	184,800	201,800	188,600	193,000	156,000	211,400	216,000	216,000
9. Total weight of engine, lb.	354,000	391,880	348,000	365,500	356,400	365,300	294,000	420,400	415,000	412,000
10. Tractive force, lb.	45,300	47,700	42,400	43,440	43,300	44,000	34,000	49,300	50,300	55,000
11. Boiler diameter, first course, inside, in.	78½	80$\frac{9}{16}$	82$\frac{7}{16}$	80⅝	78	82$\frac{7}{16}$	72 O D	86$\frac{5}{16}$	82½	88$\frac{3}{16}$
12. Boiler diameter, largest, outside, in.	90⅛	94	87⅝	91½	85⅞	93	...	91$\frac{7}{16}$	94	94
13. Length over tube sheets, ft.-in.	18-3	19-0	20-6	19-0	19-1	18-0	25-0	21-0	19-0	19-0
14. Combustion-chamber length, in.	27	36	...	43	31	42	36	...	44½	31
15. Grate area, sq. ft.	80.8	87.9	81.5	82.0	73.7	77.1	61.8	98.5	96.5	90.7
16. Tube and flue heating surface, sq. ft.‡	3,465	3,878	4,203	3,827	3,032	3,335	2,727	4,395	3,708	3,472
17. Firebox heating surface, sq. ft.‡	326	369	281	360	345	480	612	375	458	507
18. Total evaporative heating surface, sq. ft.‡	3,791	4,247	4,484	4,187	3,377	3,815	3,339	4,770	4,166	3,979
19. Superheater surface, steam side, sq. ft.	1,542	1,830	1,965	1,745	1,492	1,042	880	2,150	1,695	1,884
20. Firebox volume, cu. ft.	427	494	353	452	362	448	...	483	503	560
21. Gas area through boiler, sq. ft.	8.38	9.00	9.66	8.90	7.04	8,92	4.61	9.10	9.14	8.07
22. Type of superheater	E	E	E	E	E	A	A	E	E	E
23. Steam area through superheater, sq. in.	60.6	65.6	64.2	61.8	52.2	54.7	44.8	71.2	63.6	67.7
24. Maximum evaporation, calculated, lb. per hr.	55,960	61,710	57,770	60,500	51,600	64,000	55,480	64,780	65,400	65,740
25. Maximum evaporation, on test, lb. per hr.	...	...	84,800	...	...	...	...	...	...	...
26. Flues, number and diameter, in.	171-3½	184-3½	182-3½ 19-3½	183-3½	146-3½	48-5½	27-5½	200-3½	164-3¾	196-3½
27. Tubes, number and diameter, in.	58-2¼	62-2¼	37-2¼	59-2¼	44-2¼	199-2¼	120-2¼	46-2¼	60-2¼	8-2
28. Superheater units, number and diameter, in.	86-1$\frac{3}{16}$	93-1$\frac{3}{16}$	91-1$\frac{3}{16}$	93-1$\frac{3}{16}$	74-1$\frac{3}{16}$	48-1½	27-1¾	101-1$\frac{3}{16}$	84-1¼	102-1$\frac{3}{16}$
29. Gas area per 1,000 lb. tractive force, sq. ft.	.185	.189	.228	.206	.163	.203	.136	.185	.182	.147
30. Gas area per sq. ft. of grate, sq. ft.	.104	.103	.119	.109	.096	.116	.075	.093	.095	.089
31. Gas area per sq. ft. of tube and flue heating surface, sq. ft.	.00242	.00232	.00230	.00233	.00232	.00268	.00169	.00207	.00246	.00233
32. Grate per 1,000 lb. tractive force, sq. ft.	1.79	1.85	1.92	1.89	1.71	1.75	1.82	2.00	1.92	1.65
33. Grate per sq. ft. total evaporative heating surface, sq. ft.	.0214	.0208	.0181	.0196	.0219	.0202	.0185	.0207	.0232	.0228
34. Firebox volume per 1,000 lb. tractive force, cu. ft.	9.43	10.60	8.34	10.40	8.35	10.20	...	9.80	10.00	10.20
35. Firebox volume per sq. ft. of gas area, cu. ft.	51.0	54.9	36.6	50.8	51.4	50.2	...	53.1	55.1	69.4
36. Firebox volume per sq. ft. of grate, cu. ft.	5.29	5.61	4.34	5.51	4.91	5.81	...	4.90	5.22	6.16
37. Firebox volume per sq. ft. total evaporative heating surface, cu. ft.	113.0	116.2	78.5	108.0	107.5	117.5	...	101.1	121.0	141.0
38. Firebox heating surface per sq. ft. of grate, sq. ft.	4.03	4.20	3.45	4.40	4.68	6.23	9.90	3.81	4.75	5.59
39. Firebox heating surface per sq. ft. total evap. heating surface x 100, sq. ft.	8.60	8.70	6.25	8.60	10.2	12.60	18.35	7.85	11.0	12.75
40. Total evap. heating surface per 1,000 lb. tractive force, sq. ft.	83.6	89.0	105.9	96.3	77.8	86.6	98.1	96.8	82.9	72.4
41. Total evap. heating surface per sq. ft. of grate, sq. ft.	46.9	48.3	55.1	51.1	45.8	49.5	54.0	48.5	43.3	43.8
42. Superheat. surface per sq. ft. total evap. heat. surface, sq. ft.	.407	.431	.438	.417	.442	.274	.264	.450	.406	.474
43. Gas area through flues, per cent	84.8	85.0	80.1	85.5	86.1	51.0	42.6	89.0	86.1	98.3
44. Weight on drivers per 1,000 lb. tractive force, lb.	4,120	4,350	4,360	4,640	4,350	4,390	4,590	4,285	4,300	3,925
45. Total weight of engine per 1,000 lb. tractive force, lb.	7,810	8,200	8,210	8,425	8,245	8,300	8,640	8,550	8,250	7,490

* Streamline locomotive. † Locomotive number. ‡ Water side.

There has been a great increase in the size of locomotive boilers built in recent years in America; many of these have proved very efficient. However, much remains to be done to make possible the burning of more fuel per hour at higher combustion efficiency with lower draft loss, greater heat-absorption efficiency and higher superheat with less boiler weight per pound of steam produced.

Definition of Boiler Capacity and Existing Formulas

In designing a locomotive, the steam required for a given maximum horsepower capacity can be determined quite accurately from existing knowledge of steam consumption per i. hp.-hr., when initial and final steam conditions are specified. To this must be added the steam required for auxiliaries, train heating, air conditioning and other uses. The boiler design must be such as to deliver this maximum steam output at a superheat as high and a back pressure as low as that figured on, or else the steam consumption per i. hp. will increase and the cylinder power fall below that required. To determine the excellence of a particular boiler design and compare it with others, it is necessary to have a standard measure of comparison.

It would appear logical that the weight of steam produced per pound of total weight of the boiler in service should provide a satisfactory measure when a definite steam pressure, superheat, and draft loss are specified. It is obvious that such a yardstick is of no value, unless a standard method of predetermining the maximum evaporative capacity is found which will satisfy all conditions in a reasonable way.

In reviewing the existing methods of calculating the maximum evaporating capacity of a locomotive boiler and the over-all boiler efficiency, the designer is confronted with the fact that formulas now generally used are inadequate for predetermining the result with accuracy, particularly when large boilers are involved. In America the theory generally followed is that the

Table II—General Dimensions and Boiler Ratios of Ten-Coupled Locomotives

1. Type	2-10-0	2-10-2	2-10-2	2-10-4	2-10-4	2-10-4	2-10-4	2-10-4	2-10-4	4-10-2
2. Railroad	P.R.R.	Can.Nat.	N.Y. N.H.&H.	Cent.Ver.	C.P.R.	C.B.&Q.	C.&O.	K.C.S.	A.T.&S.F.	B.L.W.
3. Road class	I1S*	T4B	L1C	T3A	T1B	M4†	T1	J	5001	60,000
4. Date built	1922	1930	1929	1928	1938	1927	1930	1937	1938	1926
5. Boiler pressure, lb. per sq. in.	250	275	200	250	285	250	260	310	310	350
6. Cylinder diameter and stroke, in.	30½×32	24×28	30×32	27×32	25×32	31×32	29×34	27×34	30×34	27×32‡
7. Driving wheels, diameter outside tires, in.	62	57	63	60	63	64	69	70	74	63½
8. Weight on drivers, lb.	352,500	261,040	301,800	285,000	309,900	353,820	373,000	350,000	371,680	338,400
9. Total weight of engine, lb.	386,100	344,170	363,325	419,000	447,000	512,110	566,000	509,000	545,260	457,500
10. Tractive force, lb.	90,024	61,600	77,800	76,800	76,905	90,000	91,584	93,300	93,000	82,500
11. Boiler diameter, first course, inside, in.	82	74⅝	88	84½	82½	90	98	90	92¼	81⅜
12. Boiler diameter, largest, outside, in.	93	86	92 13/16	94	96½	104	108	102	104	94
13. Length over tube sheets, ft.-in.	19-1	19-3	18-0	22-0	21-0	21-6	21-0	21-0	21-0	23-0
14. Combustion-chamber length, in.	42¼	37	...	48	54	49½	66	75	72	...
15. Grate area, sq. ft.	70.0	66.7	82.0	84.4	93.5	106.5	121.7	107.0	121.5	82.5
16. Tube and flue heating surface, sq. ft.§	4,303	3,059	3,951	4,280	4,642	5,455	5,990	4,654	5,443	4,420
17. Firebox heating surface, sq. ft.§	287	347	454	423	412	449	645	500	632	342
18. Total evaporative heating surface, sq. ft.§	4,590	3,406	4,405	4,703	5,054	5,904	6,635	5,154	6,075	4,762
19. Superheater surface, steam side, sq. ft.	1,575	1,500	1,945	2,220	2,032	2,487	3,030	2,075	2,675	1,357
20. Firebox volume, cu. ft.	364	386	422	458	...	670	826	722	745	683
21. Gas area through boiler, sq. ft.	9.90	7.05	9.95	8.70	9.75	11.10	12.75	10.35	11.55	9.37
22. Type of superheater	E	E	E	E	E	E	E	E	E	A
23. Steam area through superheater, sq. in.	49.0	52.2	71.9	69.1	65.1	78.3	97.3	70.4	89.6	56.9
24. Maximum evaporation, calculated, lb. per hr.	56,660	51,750	69,000	65,110	69,120	78,520	96,210	74,680	89,140	60,410
25. Maximum evaporation, on test, lb. per hr.	65,257	...	65,750	...	...	78,710	...	...	...	69,695
26. Flues, number and diameter, in.	170-3½	146-3½	204-3½	192-3½	196-3½	222-3½	375-3½	183-3¾	249-3½	50-5½
27. Tubes, number and diameter, in.	120-2¼	44-2¼	57-2¼	33-2¼	72-2¼	87-2¼	59-2¼	73-2¼	56-2¼	206-2¼
28. Superheater units, number and diameter, in.	85-1⅛	74-1 3/16	102-1 3/16	98-1 3/16	98-1 3/16	111-1 3/16	138-1 3/16	93-1¼	127-1 3/16	50-1½
29. Gas area per 1,000 lb. tractive force, sq. ft.	.110	.114	.128	.113	.127	.123	.139	.111	.124	.114
30. Gas area per sq. ft. of grate, sq. ft.	.141	.106	.122	.103	.104	.104	.105	.097	.095	.114
31. Gas area per sq. ft. of tube and flue heating surface, sq. ft.	.00230	.00231	.00252	.00203	.00211	.00204	.00213	.00222	.00212	.00212
32. Grate per 1,000 lb. tractive force, sq. ft.	.78	1.08	1.06	1.10	1.22	1.18	1.33	1.15	1.31	1.00
33. Grate per sq. ft. total evaporative heating surface, sq. ft.	.0153	.0196	.0186	.0179	.0185	.0181	.0184	.0207	.0200	.0173
34. Firebox volume per 1,000 lb. tractive force, cu. ft.	4.04	6.26	5.42	5.96	...	7.44	9.02	7.75	8.01	8.29
35. Firebox volume per sq. ft. of gas area, cu. ft.	36.8	54.8	42.4	52.6	...	60.3	64.9	69.8	64.5	73.0
36. Firebox volume per sq. ft. of grate, cu. ft.	5.20	5.78	5.15	5.43	...	6.20	6.79	6.75	6.13	8.28
37. Firebox volume per sq. ft. total evaporative heating surface, cu. ft.	79.3	113.5	95.8	97.4	...	113.5	124.7	140.0	112.5	144.0
38. Firebox heating surface per sq. ft. of grate, sq. ft.	4.10	5.20	5.54	5.02	4.41	4.22	5.31	4.67	5.20	4.15
39. Firebox heating surface per sq. ft. total evap. heating surface x 100, sq. ft.	6.25	10.20	10.30	8.98	8.13	7.61	9.72	9.70	10.40	7.18
40. Total evap. heating surface per 1,000 lb. tractive force, sq. ft.	51.0	55.3	56.7	61.2	65.8	65.6	72.4	55.3	65.4	57.7
41. Total evap. heating surface per sq. ft. of grate, sq. ft.	65.5	51.1	53.7	55.8	54.1	55.4	54.5	48.2	50.0	57.8
42. Superheat. surface per sq. ft. total evap. heat. surface, sq. ft.	.343	.440	.441	.472	.401	.422	.457	.402	.440	.285
43. Gas area through flues, per cent	73.2	86.1	...	91.7	83.7	83.2	89.6	84.8	89.5	51.6
44. Weight on drivers per 1,000 lb. tractive force, lb.	3,920	4,240	3,870	3,710	4,020	3,925	4,070	3,775	3,985	4,100
45. Total weight of engine per 1,000 lb. tractive force, lb.	4,290	5,290	4,670	5,460	5,814	5,690	6,185	5,460	5,870	5,540

* No. 790. † No. 6320. ‡ One high-pressure and two low-pressure cylinders. § Water side.

steam-generating capacity[1] of a boiler is directly proportional to the amount of evaporating heating surface in square feet. A set of evaporative values, giving the maximum quantity of steam in pounds per square foot of heating surface per hour, generated by the firebox and flues, was prepared some years ago by F. J. Cole.[2] These values are generally used by the locomotive builders and railroads today.[1,2,3,4]

Any method of calculation that takes square feet of heating surface only into consideration must be inadequate. It is evident that the arrangement of the heating surfaces, their relationship to the grate area, furnace volume, gas area, firebox heating surface and hydraulic depth and length of the flues must be considered to give approximately correct results. The reason is that it is these relationships which determine the boiler efficiency, the back pressure required to produce the steam, and the superheat, all of which have a great influence on the efficiency and power output of the cylinders.

Strahl proposed a formula in 1913[5] which takes into consideration the size of the grate area and its ratio to the evaporating heating surface

$$WS = \frac{a}{\frac{S}{R} + 7} \times S$$

where
WS = total steam produced, lb. per hr.
R = grate area, sq. ft.
S = evaporating heating surface, sq. ft.
a = coefficient for superheated locomotive, 778 lb. per sq. ft.

[1] "Locomotive Data," The Baldwin Locomotive Works, eleventh edition, Philadelphia, Pa., 1939, pp. 21-23.

[2] "Locomotive Handbook," American Locomotive Company, Schenectady, N. Y., 1917, p. 58.

[3] "Potential Horsepower Formula Agreed to by American Locomotive Builders," Report of Federal Co-Ordinator of Transportation, Washington, D. C., Nov. 27, 1935, p. 48.

[4] "Horsepower and Tractive Effort of the Steam Locomotive," by A. I. Lipetz, Trans. A. S. M. E., vol. 55, RR-55-2, 1933, pp. 5-42.

[5] "Method of Determining the Capacity of Steam Locomotives," by Strahl, *Zeit V. D. I.*, vol. 57, 1913, pp. 251-257, 326-332, 379-386, 421-424.

Table III—General Dimensions and Boiler Ratios for Eight-Coupled Locomotives

1. Type	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4	4-8-4
2. Railroad	N.C.&St.L.	D.& R.G.W.	St.L.-S.W.	L.V.	Wabash	C.&O.	Can.Nat.	Sou.Pac.	Gt.Nor.	D.& R.G.W.
3. Road class	J2	M64	L1	T-2	O1	J3	U2D	GS2	S1	M68
4. Date built	1930	1929	1937	1931	1930	1935	1936	1936	1929	1937
5. Boiler pressure, lb. per sq. in.	250	240	250	255	250	250	250	250	250	285
6. Cylinder diameter and stroke, in.	25x30	27x30	26x30	26x32	27x32	27½x30	25½x30	27x30	28x30	26x30
7. Driving wheels, diameter outside tires, in.	70	70	70	70	70	72	73	73½	73	73
8. Weight on drivers, lb.	220,000	252,000	248,000	268,000	274,100	273,000	237,600	266,500	273,700	279,172
9. Total weight of engine, lb.	381,000	408,500	425,500	422,000	454,090	477,000	390,000	448,400	472,120	479,360
10. Tractive force, lb.	57,000	63,700	61,500	66,700	70,750	66,960	56,800	62,200	67,000	67,200
11. Boiler diameter, first course, inside, in.	79	$82\frac{15}{16}$	84½	84¼	86½	90	80⅞	84⅛	$86\frac{1}{16}$	$90\frac{3}{16}$
12. Boiler diameter, largest, outside, in.	92	96	98	98	100	100	90	96	98	100
13. Length over tube sheets, ft.-in.	20-6	22-0	20-0	21-6	21-0	21-0	21-6	21-6	22-0	21-0
14. Combustion-chamber length, in.	54	42	54	54	50	54	48½	60	52	72
15. Grate area, sq. ft.	77.3	88.0	88.3	88.3	96.2	100.0	84.3	90.4	102.0	106.0
16. Tube and flue heating surface, sq. ft.*	3,751	4,473	4,259	4,933	4,694	5,013	3,805	4,502	5,004	4,952
17. Firebox heating surface, sq. ft.*	444	446	469	508	495	525	415	350	401	555
18. Total evaporative heating surface, sq. ft.*	4,195	4,919	4,728	5,441	5,189	5,538	4,220	4,852	5,405	5,507
19. Superheater surface, steam side, sq. ft.	1,837	2,229	1,962	2,256	2,360	2,342	1,760	2,086	2,420	2,336
20. Firebox volume, cu. ft.	478	402	...	567	...	642	408	559	...	690
21. Gas area through boiler, sq. ft.	8.11	9.04	9.24	10.10	9.70	10.60	7.86	9.13	10.08	10.43
22. Type of superheater	E	E	E	E	E	E	E	E	E	E
23. Steam area through superheater, sq. in.	63.5	69.8	67.7	71.9	75.4	74.4	56.4	67.1	74.0	74.4
24. Maximum evaporation, calculated, lb. per hr.	63,040	68,280	70,100	76,500	74,780	79,640	60,810	63,575	70,570	80,800
25. Maximum evaporation, on test, lb. per hr.	...	...	...	...	...	...	...	...	...	...
26. Flues, number and diameter, in.	169-3½	195-3½	200-3½	202-3½	214-3½	220-3½	167-3½	198-3½	210-3½	222-3½
27. Tubes, number and diameter, in.	49-2¼	43-2¼	52-2¼	77-2¼	49-2¼	65-2¼	42-2¼	49-2¼	61-2¼	57-2¼
28. Superheater units, number and diameter, in.	86-$1\frac{3}{16}$	99-$1\frac{3}{16}$	102-$1\frac{3}{16}$	102-$1\frac{3}{16}$	107-$1\frac{3}{16}$	112-$1\frac{3}{16}$	85-$1\frac{3}{16}$	101-$1\frac{3}{16}$	105-$1\frac{3}{16}$	112-$1\frac{3}{16}$
29. Gas area per 1,000 lb. tractive force, sq. ft.	.142	.142	.151	.151	.137	.158	.139	.147	.150	.155
30. Gas area per sq. ft. of grate, sq. ft.	.105	.103	.105	.114	.101	.106	.093	.101	.099	.099
31. Gas area per sq. ft. of tube and flue heating surface, sq. ft.	.00216	.00202	.00217	.00205	.00207	.00211	.00207	.00203	.00201	.00211
32. Grate per 1,000 lb. tractive force, sq. ft.	1.36	1.38	1.44	1.33	1.36	1.50	1.48	1.45	1.52	1.58
33. Grate per sq. ft. total evaporative heating surface, sq. ft.	.0184	.0179	.0187	.0163	.0186	.0188	.0200	.0186	.0188	.0192
34. Firebox volume per 1,000 lb. tractive force, cu. ft.	8.38	6.31	...	8.50	...	9.60	7.15	9.00	...	10.30
35. Firebox volume per sq. ft. of gas area, cu. ft.	58.9	44.5	...	56.1	...	60.6	51.7	61.3	...	66.1
36. Firebox volume per sq. ft. of grate, cu. ft.	6.18	4.57	...	6.43	...	6.42	4.83	6.19	...	6.50
37. Firebox volume per sq. ft. total evaporative heating surface, cu. ft.	114.1	81.8	...	104.1	...	116.0	96.4	115.2	...	125.2
38. Firebox heating surface per sq. ft. of grate, sq. ft.	5.74	5.07	5.33	5.75	5.15	5.25	4.93	3.88	3.93	5.24
39. Firebox heating surface per sq. ft. total evap. heating surface x 100, sq. ft.	10.60	9.05	9.92	9.33	9.52	9.50	9.84	7.20	7.40	10.07
40. Total evap. heating surface per 1,000 lb. tractive force, sq. ft.	73.6	77.3	76.9	81.5	73.3	82.7	74.3	78.2	80.8	81.9
41. Total evap. heating surface per sq. ft. of grate, sq. ft.	54.3	55.9	53.5	61.6	54.0	55.4	50.1	53.8	53.0	51.9
42. Superheat. surface per sq. ft. total evap. heat. surface, sq. ft.	.438	.453	.415	.414	.455	.424	.417	.429	.448	.424
43. Gas area through flues, per cent	87.0	85.4	88.0	83.2	89.3	86.4	88.0	88.0	86.5	88.3
44. Weight on drivers per 1,000 lb. tractive force, lb.	3,860	3,950	4,030	4,020	3,880	4,077	4,180	4,280	4,080	4,150
45. Total weight of engine per 1,000 lb. tractive force, lb.	6,680	6,420	6,920	6,325	6,420	6,220	6,865	7,220	7,050	7,140

* Water side.

This formula, with the coefficients established by Strahl, comes close to test results on boilers with proportions similar to those which Strahl used in his analysis, but leads to unsatisfactory results on boilers with different ratios. The Strahl formula may be modified to suit different arrangements by changing the coefficients to suit, but this is not an entirely sound procedure.

Comparison of Evaporation Calculated by Different Methods with Actual Test Results

	Evaporation, lb. per hr.				
Locomotive	Strahl	Cole	Difference, per cent	Actual test	Difference, per cent
Penna. K4S No. 5341	48,000	52,150	+ 8.6	72,000	+50
Penna. M1A No. 6706	53,350*	67,850*	+27.2	99,095*	+85.7
N. Y. C. J3A	60,700*	60,500*	− 0.3	85,000*	+40

* With feedwater heater.

A method originated by Lawford H. Fry[6] for determining the maximum evaporating capacity of a boiler from the over-all boiler-efficiency curve appears to be a better approach. Fry suggested that, if the over-all boiler efficiency is plotted against pounds of coal or total heat in B.t.u. fired per sq. ft. of grate per hour, the relationship between the efficiency and rate of firing becomes a straight line. The correctness of this theory has been proved on tests examined. The fundamental equation established by Fry is

$$F = m - nG$$

where

F = boiler efficiency, per cent

G = dry coal fired, lb. per sq. ft. of grate per hr.

m = coefficient denoting theoretical efficiency at zero firing rate

n = coefficient determining slope of curve

If sufficient test data were available for all kinds of fuel and types of boilers so that the coefficients m and n or the origin and slope of the over-all boiler-efficiency curve could be predetermined with accuracy, together with the superheat and pressure at the outlet of the

[6] "A Study of the Locomotive Boiler," by Lawford H. Fry, Simmons-Boardman Publishing Corporation, New York, N. Y., 1924.

superheater, then the evaporation for any quantity of coal fired could be determined, as well as the maximum capacity.

It appears to the author that the Fry or other methods proposed do not satisfy the requirement as they stand, because draft loss and back pressure are not part of the picture. The inadequacies of both the Cole and Strahl methods are apparent when a comparison is made between the data calculated by the different methods, and the actual test results as shown in the table.

The author called attention to this some twelve years ago.[7] As far as is known, however, no tests have been made of locomotives of greatly varying boiler ratios and types of fuels to establish a satisfactory formula which may be used universally and permit evaluation of the effect that varying proportions have upon the economics of boiler performance and costs.

Boiler Tests

In its 1936 report, the A.A.R. Committee on Locomotive Construction expressed the opinion that the Cole ratios were inadequate and recommended approval

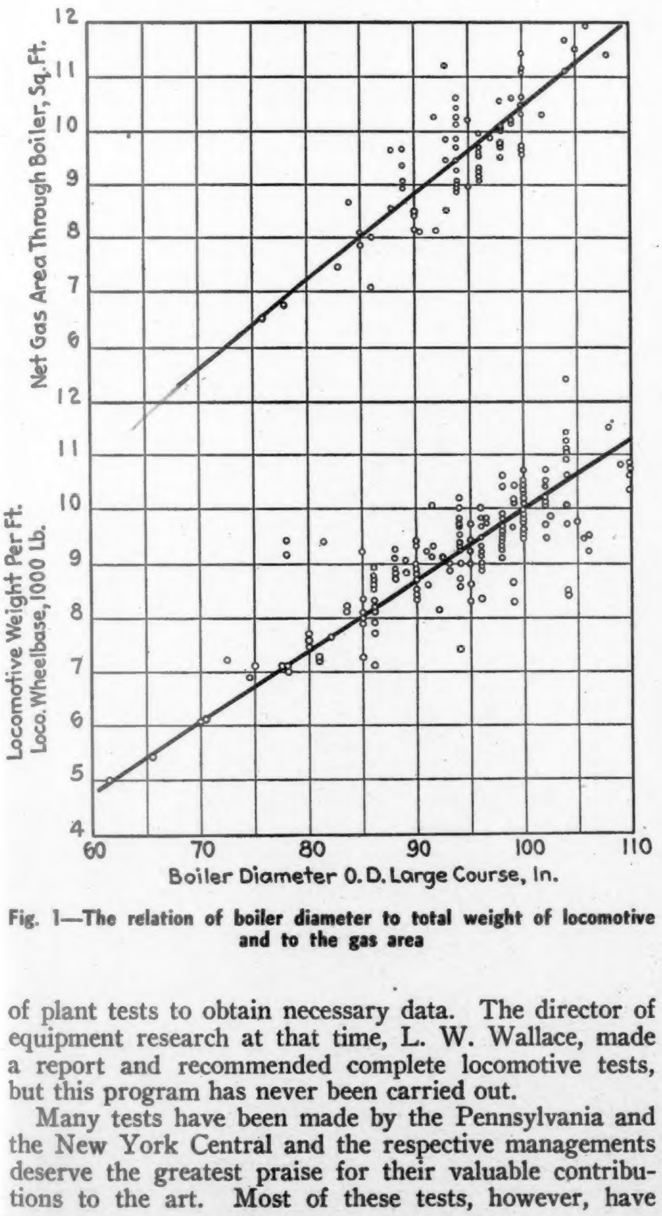


Fig. 1—The relation of boiler diameter to total weight of locomotive and to the gas area

of plant tests to obtain necessary data. The director of equipment research at that time, L. W. Wallace, made a report and recommended complete locomotive tests, but this program has never been carried out.

Many tests have been made by the Pennsylvania and the New York Central and the respective managements deserve the greatest praise for their valuable contributions to the art. Most of these tests, however, have been made on locomotives with relatively small grate areas, and with approximately the same ratios of grate areas, gas areas, firebox volumes, etc. There are few data available on the effects of very large grates and large fireboxes.

The tests recommended by the Research Division of the A.A.R. were complete and necessarily expensive. It is believed by the author that adequate information as to the efficiency and maximum capacity of boilers with large grates and furnaces can be obtained with stationary blowdown tests of some three or four boilers with widely different boiler ratios. Such tests should be conducted with oil fuel and also with several different grades of coal, sufficient to establish fundamental data now lacking. With complete data on the quantities, qualities, etc., of the fuel, water and air used, and the gases and steam produced, all losses could be segregated and closely determined.

This would permit the determination of coefficients *m* and *n* in Fry's formula. It is hoped that such a test program will be made possible as this matter is not one merely of academic importance, but is a vital item of railroad economics. The method of standing blowdown tests developed and used by the New York Central in recent years has proved very effective. This method of blowdown tests is conducted so that the steam exhausted from the cylinders through the nozzle is desuperheated to a temperature closely agreeing with that actually observed on road tests for equal capacities. The correctness of this test procedure has been proved by the fact that the front-end design and nozzle size established by such test have proved correct for best maximum performance in road service.

Gas Area in Relation to Boiler Efficiency and Capacity

It is well known that the efficiency of the boiler decreases with an increased firing rate. The rapid drop is mainly due to the high losses occurring in the form of unburned fuel escaping with the flue gases. The problem of greater fuel-burning capacity at higher efficiency of combustion is, therefore, the first item which should be considered and involves the arrangement and relative size of the gas area through the boiler, the grate area, combustion volume of the furnace, and the firebox heating surface, also, stoker construction, arrangement of firebrick arches, amount of air opening through the grates, and the introduction of secondary air above the fire bed.

In considering the matter of locomotive-boiler design, as in the case of any other structure, it is well to establish a base from which to start. This is difficult with a locomotive because the problem of its design involves a cycle of successive approximations to obtain maximum power and efficiency within the weight limitations.

In the past either the heating surface or the grate area has been the basis on which the size of the other component parts of the boiler has been determined. This is not the most logical procedure as the dimensions of these parts do not control the limit of size to which the boiler can be built, since they may be increased with the length of the boiler within considerable limits. The diameter and gas area through the boiler constitute the limiting factors because the height, width, and weight are fixed and cannot be exceeded. It is, therefore, more logical that this factor be made the basis on which the other parts are proportioned.

The capacity of the locomotive boiler is limited by its diameter because this determines the gas area of the flues through which all the gases of combustion must pass; in addition it must provide space for the super-

[7] "The Design and Proportion of Locomotive Boilers and Superheaters," by C. A. Brandt, Proceedings of the Canadian Railway Club, vol. 27, Feb., 1928, pp. 20-64.

heater through which all the steam generated must flow. It determines the flue heating surface that can be installed per unit length of flue, the area of the steam-disengaging surface and the steam volume above the water level in the boiler. This last item is of the greatest importance and is, in reality, the limiting factor in high-output operation, as evidenced by the serious difficulties experienced with water carry-over into the superheater and cylinders in many boilers.

Within the clearance, the maximum diameter of the boiler is determined by the weight limits. In order to analyze this problem, the proportions of 165 different designs of modern locomotives have been studied to ascertain the relationship between the diameter of the boiler and the total weight of the locomotive. The curve in Fig. 1 shows this relationship. It is apparent that, while there are many factors which influence the weight, the boiler diameter has been sacrificed for other features of design in many cases. If, as an example, the 104-in. boiler is noted, the total locomotive weight per foot will vary from a minimum of 8,400 lb. to a maximum of 12,400 lb., or an increase in total locomotive weight of nearly 50 per cent for the same boiler diameter. The chart indicates that there is a uniform increase of 133 lb. per ft. of total locomotive weight for every one-inch increase in the boiler diameter. This curve may facilitate studies in efforts to obtain the greatest possible diameter and gas area for allowable weights.

A relatively large gas area through the boiler in relation to the grate area and heating surface is not generally followed, as noted from a study of the ratios Nos. 29, 30 and 31, Tables I to VI.

That the gas area through the boiler is one of the most important items affecting the boiler capacity is not

Total Coal Fired, Lb. Per Hour	Total Combustion Gases, Lb. Per Hour	Velocity of Gases, Miles Per Hour			
		Section			
		A	B	C	D
8,975	99,000	135	46	35	107
23,636	158,000	263	92	70	204

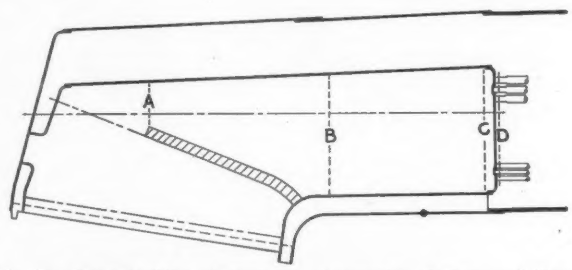


Fig. 2—Gas velocities through the locomotive firebox and flues

generally appreciated, but may be better understood if actual test figures are analyzed. From the test of the Pennsylvania M1A, Locomotive No. 6706, on which sufficient data are available to permit determination of the weight of the gases passing through the boiler, the data shown in Fig. 2 have been compiled. It is interesting to note that, at high firing rates, the gas velocity through the tubes near the back tube sheet reaches the high figure of 204 miles per hour. The great importance of eliminating unnecessary restrictions in the gas passages and the superheater units is apparent.

As the gas area is increased, the velocity and frictional resistance of the gases passing into and through the boiler flues are lowered. This will reduce the draft requirement and in turn the cylinder back pressure, en-

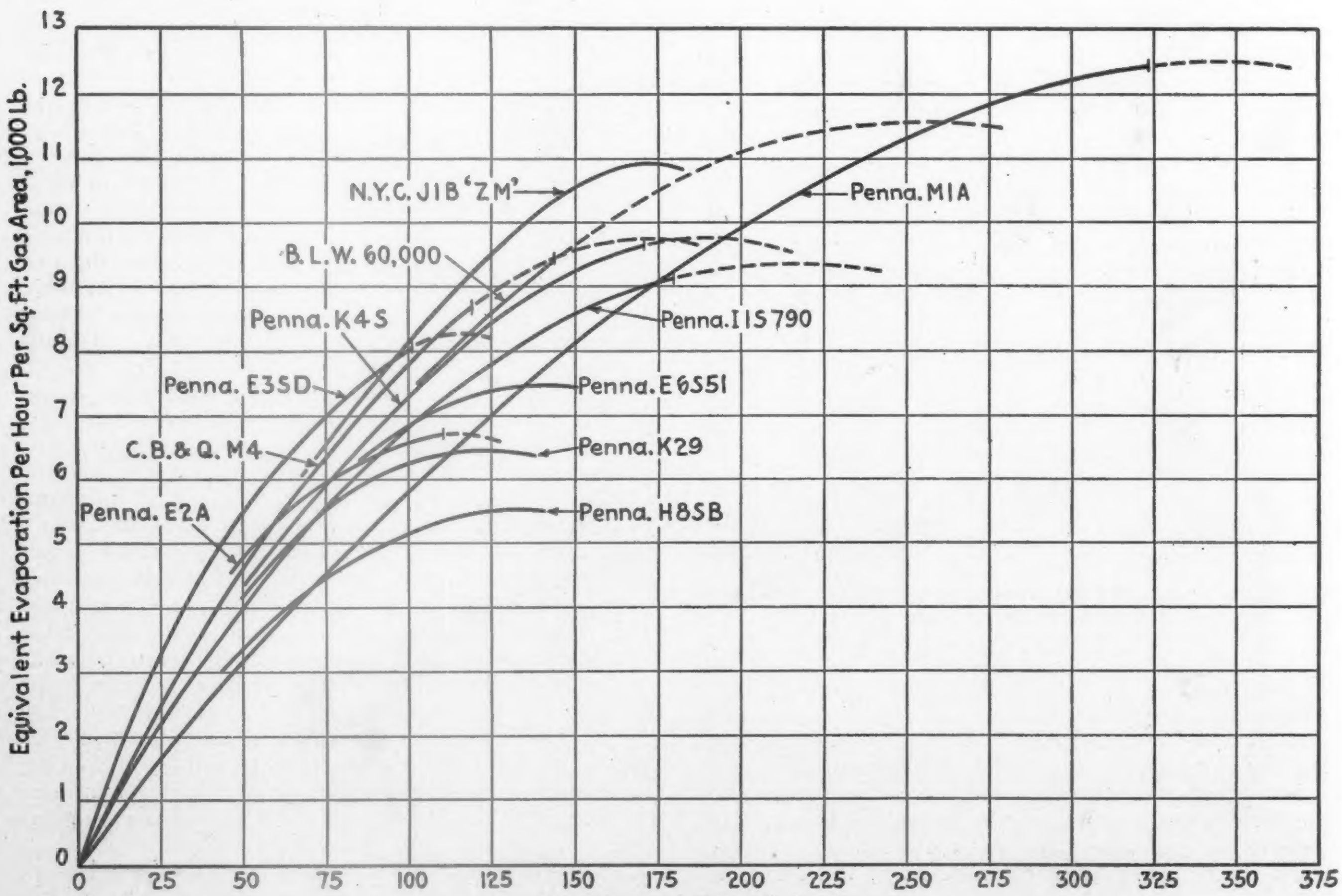


Fig. 3—Equivalent evaporation per sq. ft. of gas area versus coal per sq. ft. of grate

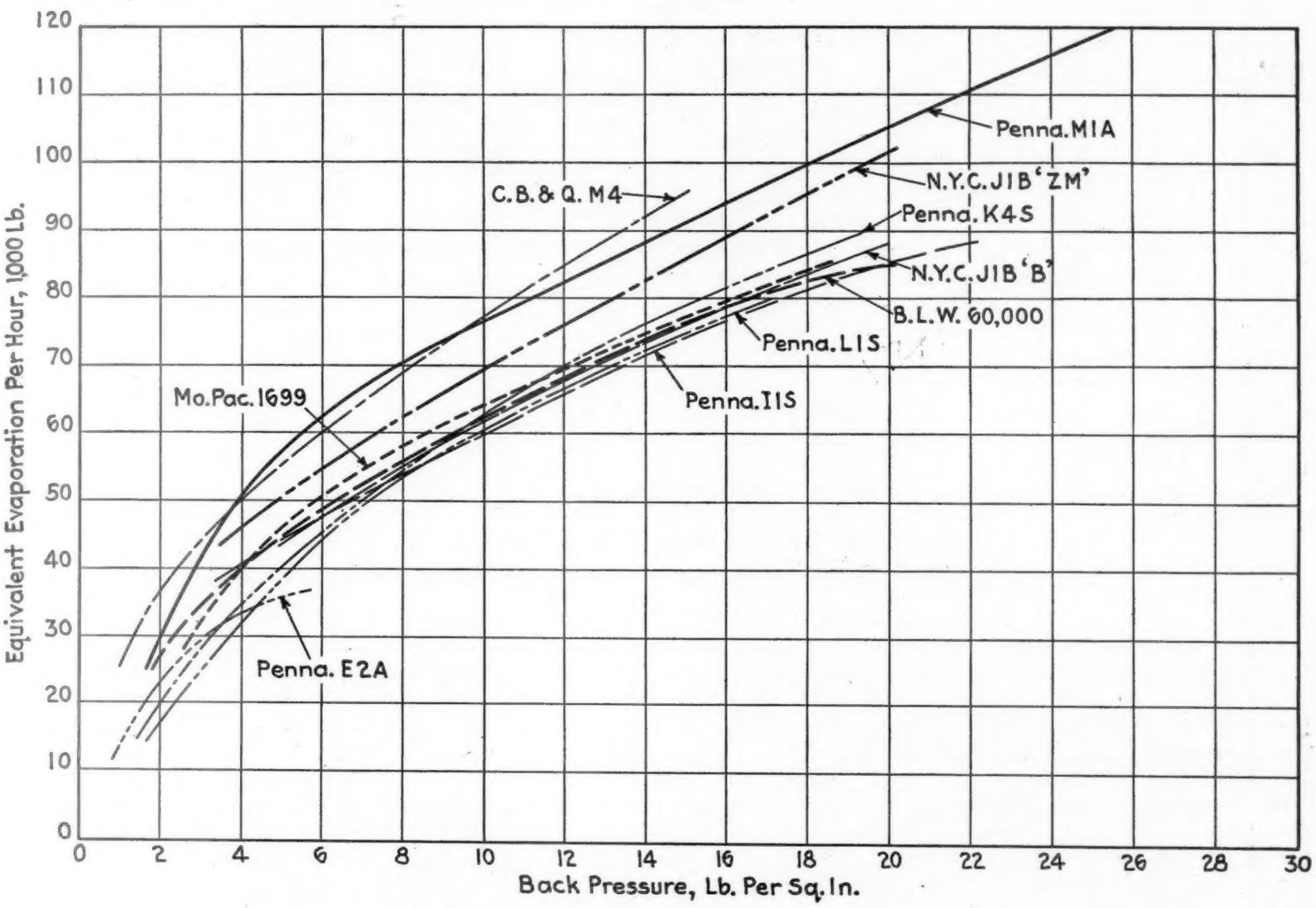


Fig. 4—Equivalent evaporation versus back pressure

larging the power output of the locomotive in direct proportion to the resultant increase in the m.e.p. This is important when high power output is desired at high speed as the reduction of 1 lb. of cylinder back pressure gives as much increase in power as a 4-lb. gain in pressure on the steam-inlet or admission side.

The importance of a large gas area through the boiler was recognized years ago. C. D. Young[8] pointed out in 1915 that from his experience the maximum evaporative capacity of a boiler was in direct ratio to the total gas area through the flues and was apparently limited to an amount 7,000 lb. actual or 9,100 lb. equivalent evaporation per hr. per sq. ft. of gas area for the engines tested up to that time. This has been bettered considerably on modern locomotives with type-E superheaters and more powerful draft arrangements to about 11,000 lb. equivalent evaporation per sq. ft. of gas area. To illustrate the limiting effect that the gas area has upon the evaporative capacity of a locomotive boiler, the test results from several locomotives have been plotted in Fig. 3. The relation between equivalent evaporation and back pressure is illustrated in Fig. 4.

A study of the relationship between the gas area and the boiler diameter of a number of modern boilers has been plotted in Fig. 1 and it will be noted that full advantage has not been taken in many cases in providing the largest possible gas area. For some boiler diameters the gas area is as much as 31½ per cent smaller than the maximum possible. This is perhaps due to a difference of opinion as to the most suitable distance between the top of the crown sheet and the inside of the boiler shell at the top; as to the water space between the combustion chamber and the boiler shell, or the clearance between tube holes and tube-sheet flanges; and as to tube spacing. The type of combustion chamber, the thickness of the flues and the superheater design also influence the gas area. Offhand, it may seem that these details are not of great consequence, but really they are very important.

The distance between the top of the crown sheet and the boiler shell determines the steam space and steam-disengaging surface. It is obvious that the smaller this distance is made the greater will be the tube-sheet area. Information as to the most satisfactory relationship between the height of the crown sheet and other dimensions of the boiler is lacking, but this question deserves most careful consideration.

A slight increase in the water space between the combustion chamber and boiler shell decreases the available gas area considerably. On some boilers, this water space has been made as large as 8 in., while very many large boilers are operating successfully with a water space of only 5 in., giving a large percentage increase in gas area. It appears that the smaller water space imparts a higher velocity to the water circulation at this point, minimizing mud collection which the large water space was supposed to eliminate.

Firebox Volume in Relation to Combustion Efficiency

Earlier in this paper, it was pointed out that the firebox volume is not subject to such limitations as the gas area. There are limitations to the width and height of the firebox which are important, but the length of the grate and firebox may be extended considerably beyond present general practice. This is fortunate as the most

[8] Test Bulletin, No. 28, Pennsylvania Railroad, 1915, p. 35.

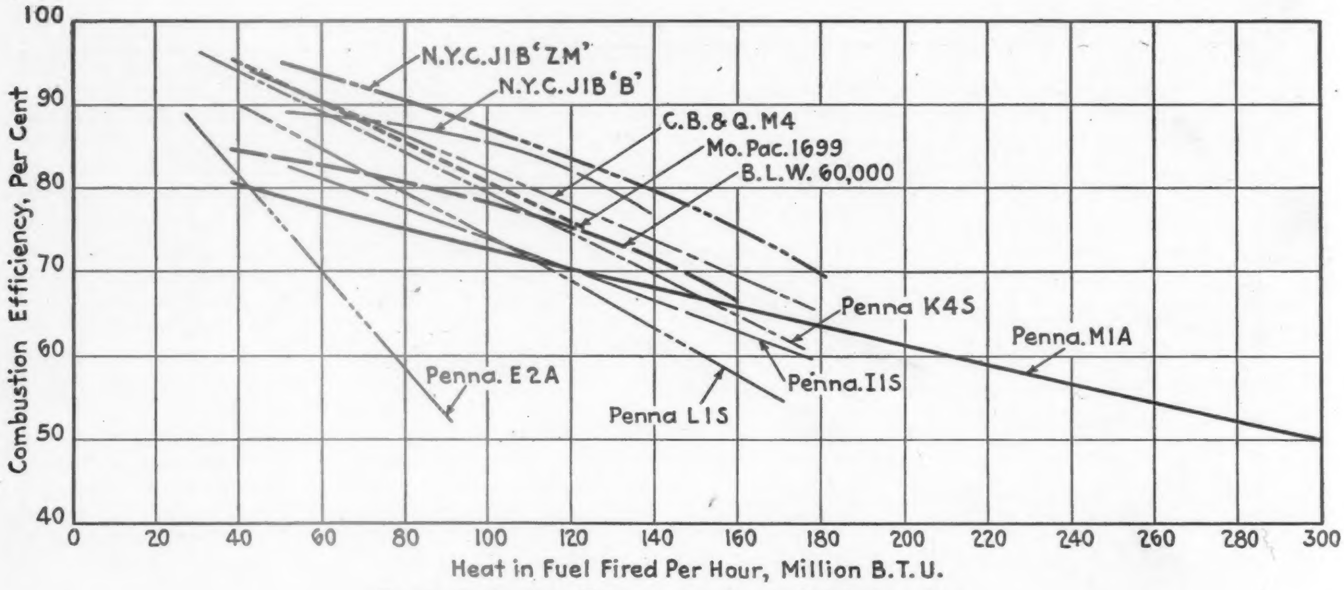


Fig. 5—Combustion efficiency versus heat in fuel fired per hour

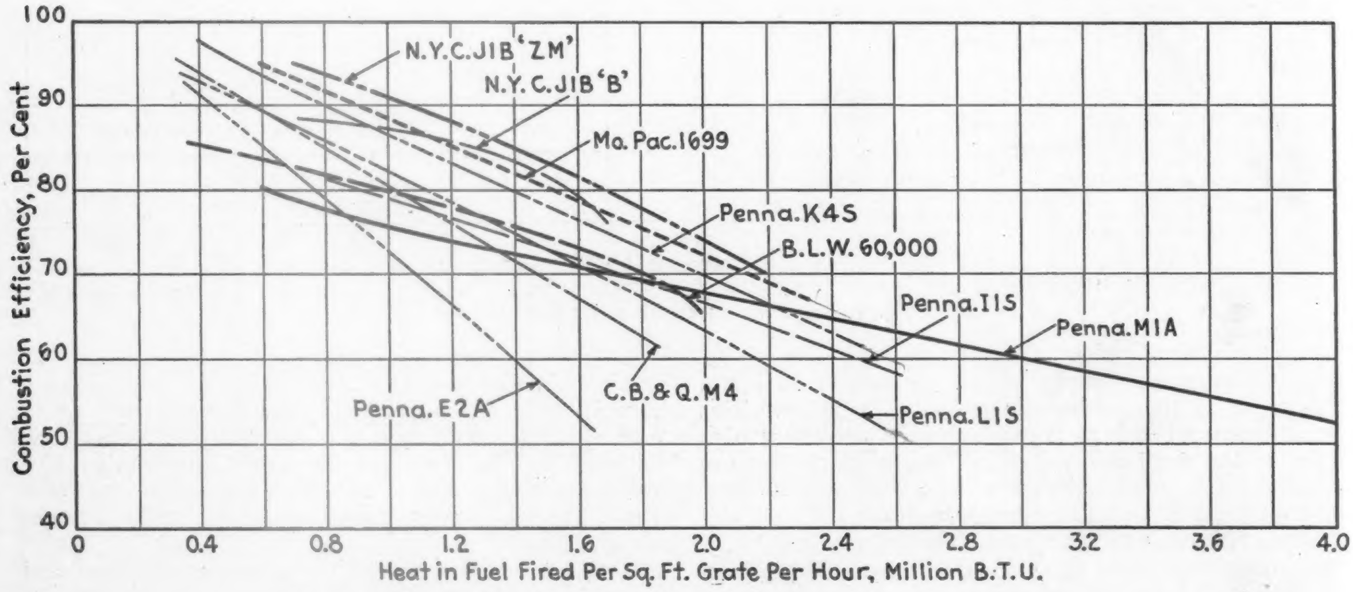


Fig. 6—Combustion efficiency versus heat in fuel fired per sq. ft. of grate per hour

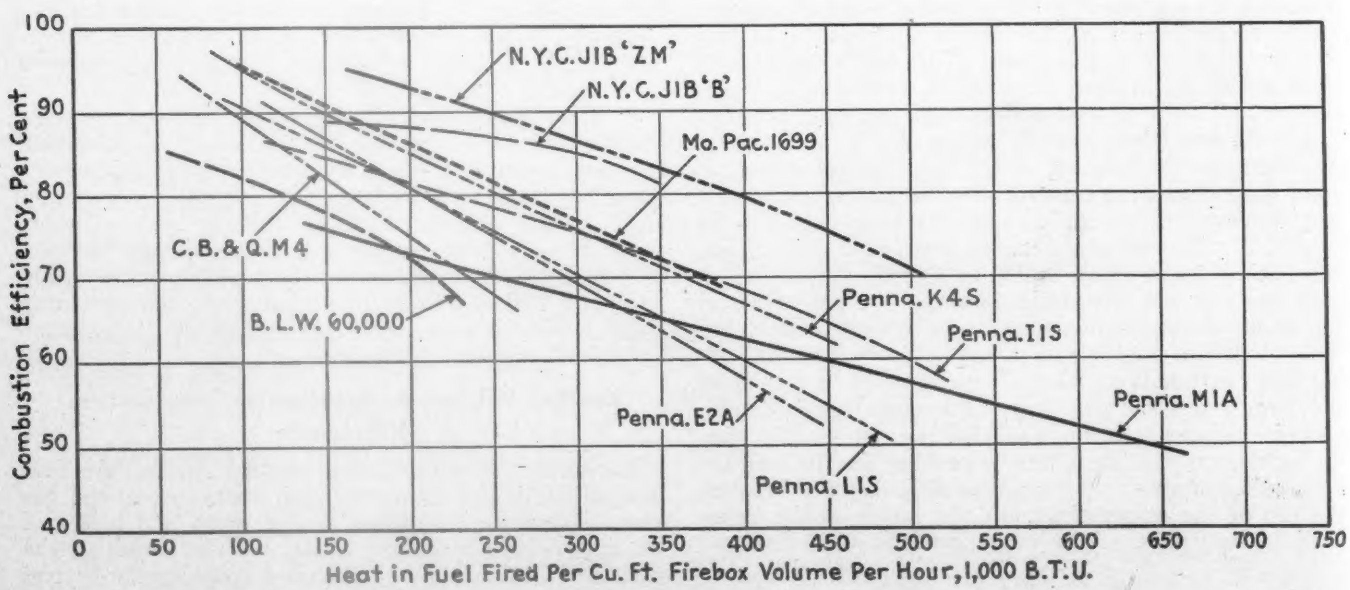


Fig. 7—Combustion efficiency versus heat in fuel fired per cu. ft. of firebox volume per hour

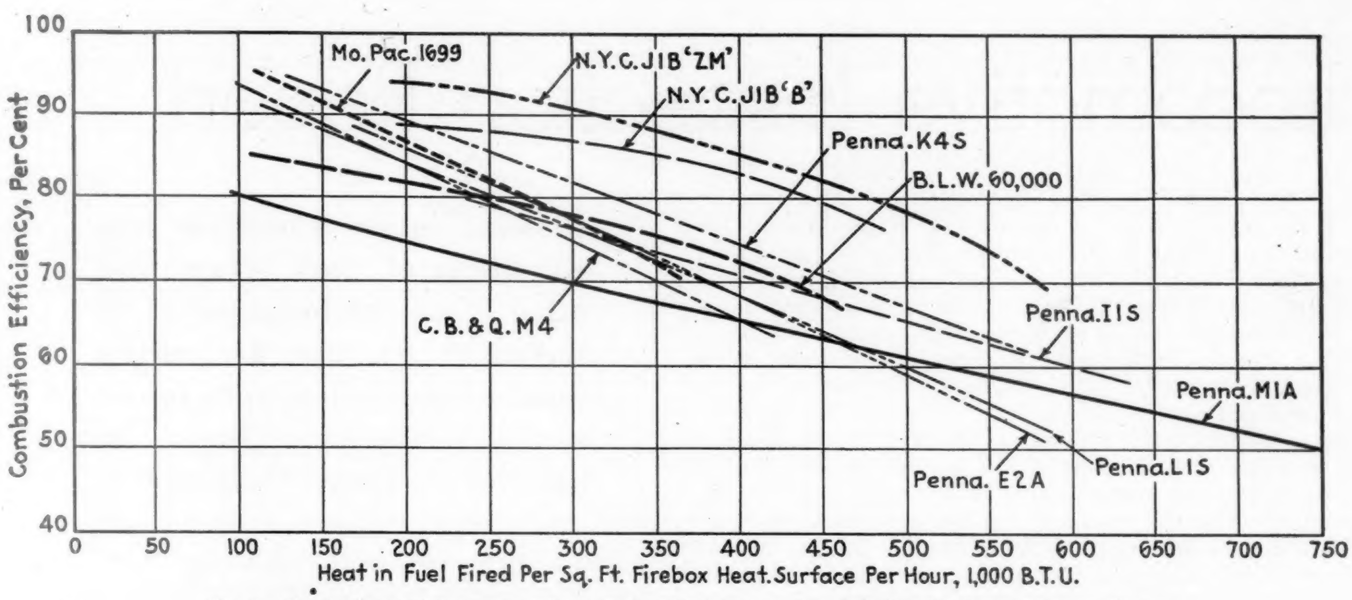


Fig. 8—Combustion efficiency versus heat in fuel fired per sq. ft. of firebox heating surface per hour

serious matter confronting locomotive designers today is the problem of improving the efficiency of combustion in the firebox, particularly at high-capacity operation. Incomplete combustion of fuel causes heavy losses in unburned fuel, high maintenance cost due to cinder cutting of firebox sheets, stay-bolt heads, tubes, and superheater. Damage due to fires along the right of way set by sparks, loss in good will to the general public due to smoke near railroad terminals, and loss of passenger traffic for the same cause constitute a serious challenge to designers and operators of steam locomotives.

Improvement in the combustion process of the locomotive furnace is a problem of primary importance and the proportions of the furnace should be such as not only to make possible complete combustion of the fuel with the elimination of smoke and cinders, but also to absorb sufficient radiant heat to control the furnace gases to a temperature low enough to prevent slagging at the flue sheet and on the superheater units. One would assume that sufficient data were available today to determine the correct volume of the firebox in relation to the grate area and heating surface. A study of Tables I to VI indicates, however, that there is little agreement on this subject; great variations exist in these basic ratios. To cite an example: The important ratio of firebox volume to grate area ranges from a minimum of 3.70 to a maximum of 8.28 cu. ft. per sq. ft. of grate area. Compared on the basis of tractive force the furnace volume varies from a minimum of 4.04 to a maximum of 11.67 cu. ft. per 1,000 lb. of tractive force.

The difference in opinion among locomotive designers as to the correct size of firebox and combustion chamber is due to the absence of pertinent facts and test data to permit a scientific evaluation of the various factors. It is known that the first cost of the heating surfaces in the firebox and combustion chamber is very high as is also the cost of the maintenance and inspection when compared with the tube heating surface. What the relative cost of firebox and tube maintenance amounts to is unknown. Little information is available as to the cost of boiler maintenance, since few American railroads keep accounts to permit such a study. In the report of the Federal Co-Ordinator of Transportation, November, 1935, the estimated average cost of boiler maintenance is given as 30 per cent of the total cost of locomotive repairs. The firebox maintenance is the greatest part of this cost. As a large modern boiler will have from 5,000 to 8,000 staybolts in the firebox which must be tested at regular intervals, the size of the firebox is a very important item when the question of intensive utilization of the locomotive is considered.

Against these items of cost should be balanced the great improvements in boiler efficiency and capacity which are generally believed to be obtained by the use of large fireboxes. The heat absorption of the firebox heating surfaces is from six to ten times as great as the average tube heating surface. Another important point not usually considered is that the radiant-heat absorption by the firebox is accomplished without the expenditure of any energy, while the heat absorption by convection in the tubes consumes considerable power to create the high draft required. The energy required to move the gases increases with the fourth power of the weight velocity. Only a few tests have been made which supply data permitting the calculation of combustion efficiency. These tests have been mainly on engines, the boiler ratios of which are very much alike, and thus the effect of greatly different boiler ratios is not clearly discernible from available data. To illustrate the effect which different boiler ratios have upon combustion efficiency, the data from all tests available have been plotted in Figs. 5 to 8.

It is, however, becoming more generally recognized that the larger the furnace volume of a locomotive, the greater is the opportunity for completing the processes of combustion. This is true only if the larger furnace volume is obtained in conjunction with maximum boiler diameter, furnace width and depth, and not by furnace length alone.

An examination of a typical boiler diagram, Fig. 2, shows the gas velocities at several points in the firebox, an explanation being given why the largest possible cross-sectional area through the firebox is very important. The velocity of the furnace gases in this case between the top of the arch and the crown sheet is as high as 260 m.p.h. It is remarkable that the small particles of coal being blown by the stoker jets into the center of a hurricane of such velocity are able to settle on the grate or have time to burn at all.

A particle of fine coal caught in the gas stream will

(*Continued on page 64*)

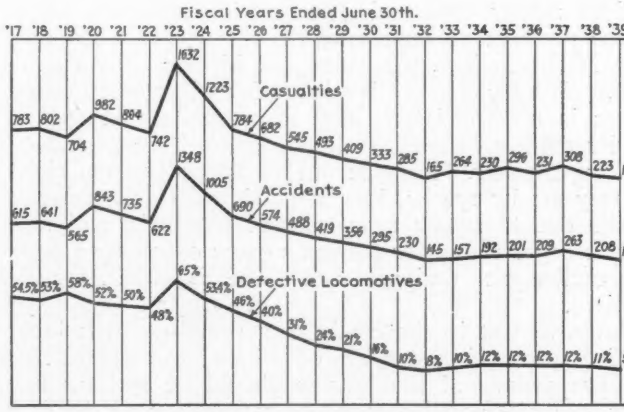
Locomotive Inspection Report

General improvement in the condition of both steam and other types of locomotives is indicated by the decreasing number of defective units found while, at the same time, the number of inspections increases

THE annual report of the Bureau of Locomotive Inspection, Interstate Commerce Commission, submitted by John M. Hall, chief inspector, covering the fiscal year ended June 30, 1939, shows an improvement in the condition of locomotives, both steam and other than steam. In the case of steam locomotives, there was a decrease of 1,432 in the number for which reports were filed; an increase of 420 in the number of locomotives inspected; a decrease of 1,951 in the number found defective; a decrease of 2 per cent in the number inspected and found defective; a decrease of 211 in the number ordered out of service, and a decrease of 8,724 in the total number of defects found. In the case of locomotives other than steam, the report shows increases in the number of locomotive units for which reports were filed and in the number inspected; decreases are shown for the number found defective, the total number of defects found and the percentage inspected and found defective.

The accompanying chart shows the percentage of defective locomotives, the number of accidents and the number of casualties for the fiscal years ended June 30, 1917, to 1939, inclusive. Summaries and tables included in the report show separately accidents and other data in connection with steam locomotives and tenders and their appurtenances and similar data covering locomotives other than steam.

During the fiscal year ended June 30, 1939, the number of steam locomotives for which reports were filed totaled 45,965. The number of steam locomotives inspected totaled 105,606 and 9,099, or 9 per cent, were found defective, and 468 were ordered out of service. In 1938 105,186 steam locomotives were inspected, of which 11,050, or 11 per cent, were found defective, and 679 ordered out of service. In the year ended June 30, 1937, a total of 100,033 steam locomotives were inspected, 12,402, or 12 per cent, were found defective, and 852 were ordered out of service. The total number of defects found and shown in the last three reports were: 33,490 in 1939, 42,214 in 1938, and 49,746 in 1937. In the case of locomotives other than steam, reports were filed covering 2,716 locomotives in the year ended June 30, 1939. The inspections made totaled 4,581, of which 260, or 6 per cent, were found defective, and 14 ordered out of service. The total number of defects found on this type of motive power in that year totaled 696. These figures compare with the following for the previous year



A 23-year record of accidents, casualties and defective locomotives

Result of an explosion caused by the overheating of the crown sheet due to low water—The boiler landed 318 ft. from the point where the explosion occurred and wreckage was scattered to distances up to 2,165 ft.—Two employees were killed

Table I—Number of Casualties Classified According to Occupation—Steam Locomotive Accidents

	Year ended June 30									
	1939		1938		1937		1936		1935	
	Killed	Injured	Killed	Injured	Killed	Injured	Killed	Injured	Killed	Injured
Members of train crews:										
Engineers	4	46	3	70	8	106	4	75	7	65
Firemen	6	66	2	80	5	78	6	72	4	70
Brakemen	2	18	..	31	3	30	3	28	2	26
Conductors	..	5	..	6	1	18	..	13	..	10
Switchmen	..	6	..	7	..	10	..	2	..	3
Roundhouse and shop employees:										
Boilermakers	1	1	..	2	2	2	..	..	..	6
Machinists	..	2	..	..	..	2	..	4	1	3
Foremen	..	..	..	1	..	..	..	3	..	2
Inspectors	..	..	..	1	..	..	..	2	..	1
Watchmen	..	1	2	..	1	1	1	1	1	1
Boiler washers	..	..	..	1	..	..	..	..	..	..
Hostlers	..	1	..	6	..	9	..	3	..	3
Other roundhouse and shop employees	..	2	..	1	..	3	..	3	..	6
Other employees	..	2	..	3	1	14	..	5	14	49
Nonemployees	2	14	..	7	4	10	2	4	..	22
Total	15	164	7	216	25	283	16	215	29	267

Interior of a firebox after a crown sheet failure due to low water—In this accident, in which two employees were injured, the smokebox door and front were blown off and forced through the vestibule and partially through the door of an unoccupied sleeping car

ending June 30, 1938: Total number inspected, 4,024; number found defective, 274; percentage inspected and found defective, 7; number ordered out of service, 9; total number of defects found, 769.

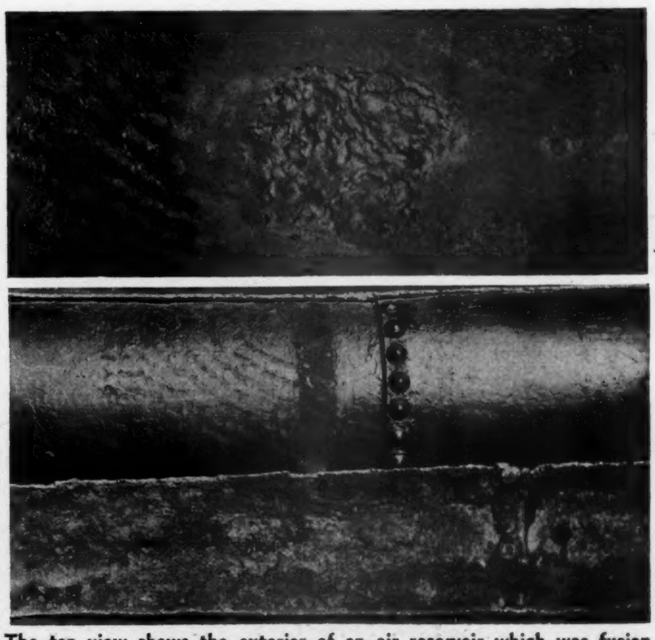
Table I shows the number of casualties in connection with steam locomotive accidents classified according to occupation; Table II shows the nature of the defects found on steam locomotives and gives the data on inspections and reports. Similar data for locomotives other than steam will be found in Table III.

Boiler Explosions

Six boiler explosions occurred during the past fiscal year and are commented on in the report. All of these explosions, in which 12 persons were killed and 11 were injured, were caused by the overheating of crown sheets due to low water.

Extension of Time for Removal of Flues

A total of 1,009 applications were filed for extensions of time for removal of flues, as provided in Rule 10. The investigations of the bureau disclosed that in 64 of these cases the condition of the locomotives was such that extensions could not properly be granted. In the case of 31 locomotives, conditions were such that full extensions requested could not be authorized, but extensions for shorter periods of time were allowed. Requests for 56 extensions were granted after defects disclosed by the



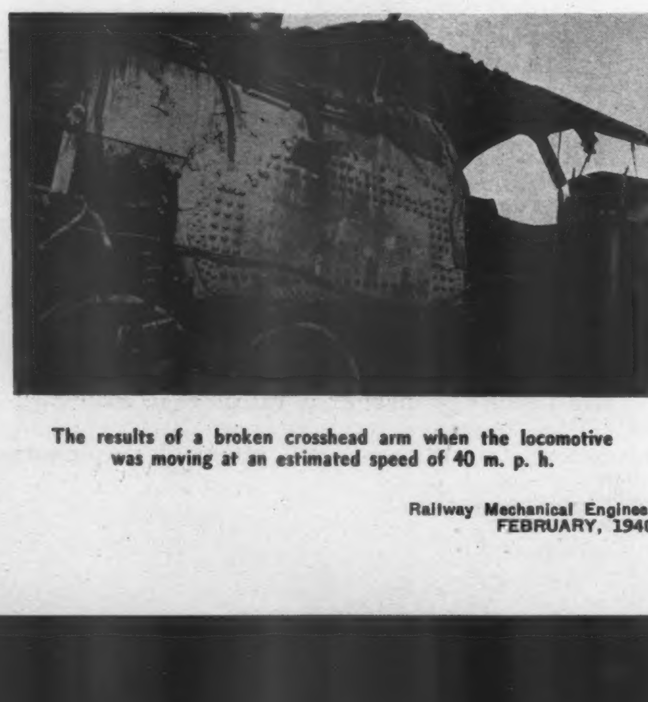
The top view shows the exterior of an air reservoir which was fusion welded where metal had deteriorated and wasted through along the bottom—The lower illustration shows the condition of the inside of the same reservoir along the bottom

Table II—Number of Steam Locomotives Reported, Inspected, Found Defective, and Ordered from Service

Parts defective, inoperative or missing, or in violation of rules	Year ended June 30—					
	1939	1938	1937	1936	1935	1934
1. Air compressors	518	689	766	740	733	660
2. Arch tubes	28	66	105	74	74	127
3. Ashpans and mechanism	67	72	80	79	94	87
4. Axles	2	13	10	13	10	6
5. Blow-off cocks	204	226	199	236	283	289
6. Boiler checks	279	301	382	356	413	407
7. Boiler shell	272	331	347	383	396	372
8. Brake equipment	1,577	2,044	2,322	2,480	2,449	2,326
9. Cabs, cab windows, and curtains	943	1,226	1,807	1,638	1,273	1,342
10. Cab aprons and decks	260	326	466	450	368	343
11. Cab cards	92	109	145	166	142	129
12. Coupling and uncoupling devices	60	73	74	65	73	54
13. Crossheads, guides, pistons, and piston rods	739	905	1,160	1,056	1,086	1,100
14. Crown bolts	47	59	76	63	75	77
15. Cylinders, saddles and steam chests	1,232	1,645	2,206	1,717	1,547	1,491
16. Cylinder cocks and rigging	418	585	729	605	627	654
17. Domes and dome caps	90	109	101	114	94	105
18. Draft gear	450	740	522	513	423	401
19. Draw gear	360	479	560	451	414	480
20. Driving boxes, shoes, wedges, pedestals and braces	1,330	1,688	1,637	1,712	1,573	1,472
21. Firebox sheets	238	244	371	295	343	356
22. Flues	165	159	225	178	173	203
23. Frames, tail pieces and braces, locomotive	708	1,001	1,053	997	1,006	951
24. Frames, tender	71	131	120	113	124	128
25. Gages and gage fittings, air	155	230	261	257	275	212
26. Gages and gage fittings, steam	226	279	324	350	320	289
27. Gage cocks	361	451	538	579	480	384
28. Grate shakers and fire doors	252	403	470	400	394	404
29. Handholds	349	405	510	502	464	377
30. Injectors, inoperative	26	26	38	40	39	33
31. Injectors and connections	1,457	1,784	2,020	2,085	2,035	1,909
32. Inspections and tests not made as required	6,645	8,204	9,638	9,005	8,344	8,173
33. Lateral motion	243	325	446	404	389	351
34. Lights, cab and classification	50	48	90	78	81	79
35. Lights, headlight	177	257	313	251	257	218
36. Lubricators and shields	200	212	254	255	191	215
37. Mud rings	248	203	272	237	241	247
38. Packing nuts	408	448	487	508	527	491
39. Packing, piston rod and valve stem	739	913	1,393	1,133	906	833
40. Pilots and pilot beams	104	154	133	178	152	174
41. Plugs and studs	179	238	238	236	167	242
42. Reversing gear	317	404	492	463	414	390
43. Rods, main and side, crank pins and collars	1,293	1,669	2,348	2,093	1,826	1,670
44. Safety valves	97	125	132	125	100	108
45. Sanders	432	536	655	678	779	697
46. Springs and spring rigging	2,340	2,901	3,172	3,008	2,765	2,854
47. Squirt hose	75	94	133	134	113	107
48. Stay bolts	181	211	276	279	240	285
49. Stay bolts, broken	258	380	542	520	512	455
50. Steam pipes	285	410	446	526	463	489
51. Steam valves	115	141	165	227	212	267
52. Steps	490	631	678	615	640	567
53. Tanks and tank valves	837	955	1,009	877	913	862
54. Telltale holes	58	67	79	127	102	93
55. Throttle and throttle rigging	638	685	909	760	733	639
56. Trucks, engine and trailing	628	762	785	861	811	898
57. Trucks, tender	665	907	1,018	1,108	1,120	918
58. Valve motion	554	722	798	824	799	784
59. Washout plugs	487	626	598	714	679	776
60. Train-control equipment	5	11	12	6	4	8
61. Water glasses, fittings and shields	690	915	1,049	1,118	951	907
62. Wheels	466	577	803	790	697	734
63. Miscellaneous—Signal appliances, badge plates, brakes (hand)	610	684	759	608	563	572
Total number of defects	33,490	42,214	49,746	47,453	44,491	43,271
Locomotives reported	45,965	47,397	48,025	49,322	51,283	54,283
Locomotives inspected	105,606	105,186	100,033	97,329	94,151	89,716
Locomotives defective	9,099	11,050	12,402	11,526	11,071	10,713
Percentage of inspected found defective	9	11	12	12	12	12
Locomotives ordered out of service	468	679	934	852	921	754

Table III—Number of Locomotives Other Than Steam Reported, Inspected, Found Defective and Ordered from Service

Parts defective, inoperative or missing, or in violation of rules	Year ended June 30—					
	1939	1938	1937	1936	1935	1934
Air compressors	14	6	6	2	5	3
Axles, truck and driving	1	5	4	6	1	..
Batteries	1	1	4	..	7	..
Boilers	6	6	5	5	3	1
Brake equipment	50	74	97	66	46	15
Cabs and cab windows	36	25	51	30	33	9
Cab cards	18	11	25	..	..	..
Cab floors, aprons, and deck plates	13	8	17	10	6	1
Controllers, relays, circuit breakers, magnet valves, and switch groups	13	7	8	..	..	5
Coupling and uncoupling devices	4	4	3	..	..	..
Current-collecting apparatus	5	8	4	16	3	3
Draft gear	17	23	28	24	21	8
Draw gear	4	3	1	1	..	..
Driving boxes, shoes and wedges	52	16	14	5	5	7
Frames or frame braces	9	37	5	15	4	6
Fuel system	35	47	152	44	15	4
Gages or fittings, air	6	11	1	6	4	..
Gears and pinions	2	2	2	..	..	1
Handholds	8	13	11	8	3	..
Inspections or tests not made as required	185	204	237	186	124	52
Insulation and safety devices	4	13	13	20	15	2
Internal-combustion engine defects, parts and appliances	32	26	50	23	4	4
Jack shafts	6	1	..	1	..	..
Jumpers and cable connectors	1	1	2	..	..	..
Lateral motion, wheels	1	..	1	2	..	3
Lights, cab and classification	3	2	5	6	1	..
Lights, headlight	4	4	11	4	2	..
Meters, volt and ampere	2	2	1	2	..	..
Motors and generators	19	18	10	14	5	4
Pilots and pilot beams	6	1	7	6	5	..
Plugs and studs	..	..	1	..	..	..
Quills	7	6	3	..	..	..
Rods, main, side, and drive shafts	2	2	23	2	10	4
Sanders	28	37	52	25	21	2
Springs and spring rigging, driving and truck	16	43	36	29	20	4
Steam pipes	..	5	1	2	..	..
Steps, footboards, etc.	18	23	13	..	..	..
Switches, hand-operated, and fuses	5	7	2	2	2	1
Transformers, resistors, and rheostats	1	3	..	..	1	1
Trucks	33	40	41	42	46	3
Water tanks	1	..	1	..	..	..
Water glasses, fittings, and shields	1	3	..	4	6	..
Warning signal appliances	1	3	2	1	..	..
Wheels	16	11	21	26	6	8
Miscellaneous	10	7	20	39	25	7
Total defects	696	769	991	674	449	158
Locomotive units reported	2,716	2,555	2,416	2,361	1,911	1,288
Locomotive units inspected	4,581	4,024	3,615	3,118	1,620	1,436
Locomotive units defective	260	274	328	252	146	69
Percentage inspected found defective	6	7	9	8	9	5
Locomotive units ordered out of service	14	9	24	11	5	4



The results of a broken crosshead arm when the locomotive was moving at an estimated speed of 40 m. p. h.

Two cases of scale accumulation—In the boiler check (above) the diameter of the opening was reduced from 2½ in. to 1 in. causing erratic operation of the injector—The lower illustration shows a water column practically closed by hard scale

The condition of a driving-brake pull rod and pins on a locomotive withheld from service by a Bureau inspector—The report of a monthly inspection by the railroad, two days previous to the Bureau's inspection, showed the condition of the brake equipment as "good"

bureau's investigations were repaired. Fifteen applications were cancelled for various reasons. Extensions for the full time requested were granted in 843 cases.

Locomotives Propelled by Power Other Than Steam

There was an increase of one in the number of accidents occurring in connection with locomotives other than steam and an increase of one in the number of persons injured as compared with the previous year. No deaths occurred in either year.

During the year 6 per cent of the locomotives inspected were found with defects or errors in inspection that should have been corrected before the locomotives were put into use as compared with 7 per cent in the previous year. There was an increase of 5 in the number of locomotives ordered withheld from service by the inspectors, because of the presence of defects that rendered the locomotives immediately unsafe.

Specification Cards and Alteration Reports

Under Rule 54, of the Rules and Instructions for Inspection and Testing of Steam Locomotives, 131 specification cards and 4,493 alteration reports were filed, checked, and analyzed. Corrective measures were taken with respect to numerous discrepancies found as a result of checking these reports.

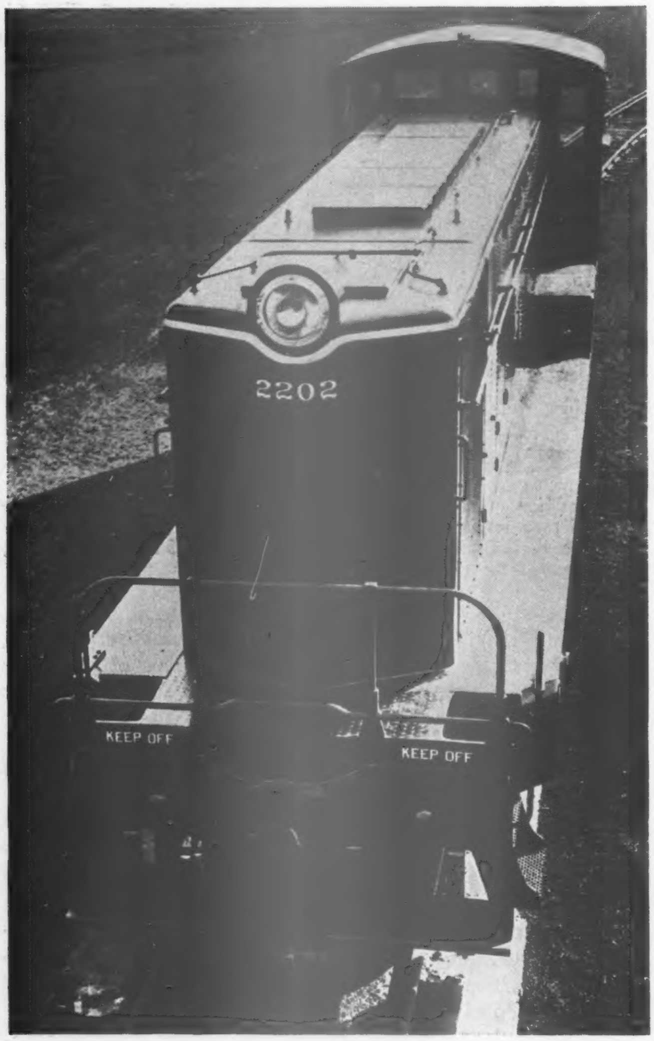
Under Rules 328 and 329 of the Rules and Instructions for Inspection and Testing of Locomotives Other Than Steam, 252 specifications and 90 alteration reports were filed for locomotive units and 60 specifications and 36 alteration reports were filed for boilers mounted on locomotives other than steam. These were checked and analyzed and corrective measures taken with respect to discrepancies found.

No formal appeal by any carrier was taken from the decisions of any inspector during the year.

This crown sheet failure resulted in the death of two employees

Baldwin Diesel-Electric Switchers

Stock locomotives built at the Eddystone plant are powered with eight- and six-cylinder, 1,000- and 660-hp. De La Vergne Diesel engines and Westinghouse electric generating, traction and control equipment



THE first of a group of 28 Diesel-electric switching locomotives of 1,000 and 660 hp. to be built in line production at the Eddystone (Pa.) plant of the Baldwin Locomotive Works, have recently been completed and five of the 1,000 hp. type were delivered to the Atchison, Topeka & Santa Fe.

Each locomotive is powered with a single De La Vergne, Model VO, Diesel engine of eight or six cylinders, respectively, and Westinghouse generating, control and traction equipment. The eight-cylinder, 1,000-hp. locomotive weighs 120 tons, while the six-cylinder, 660-hp. unit weighs 100 tons. The comparative specifications are shown in the table.

The 660-hp. locomotive has a continuous rating of 28,000 lb. tractive force at 6.5 m.p.h. and the 1,000-hp. unit a continuous rating of 33,600 lb. tractive force at 8.3 m.p.h. A tractive force above 33,600 lb. can be developed by the 1,000 hp. locomotive (above 28,000 lb. by the 660-hp. locomotive) for shorter periods, depending upon the initial temperature of the traction motors.

The De La Vergne Engines

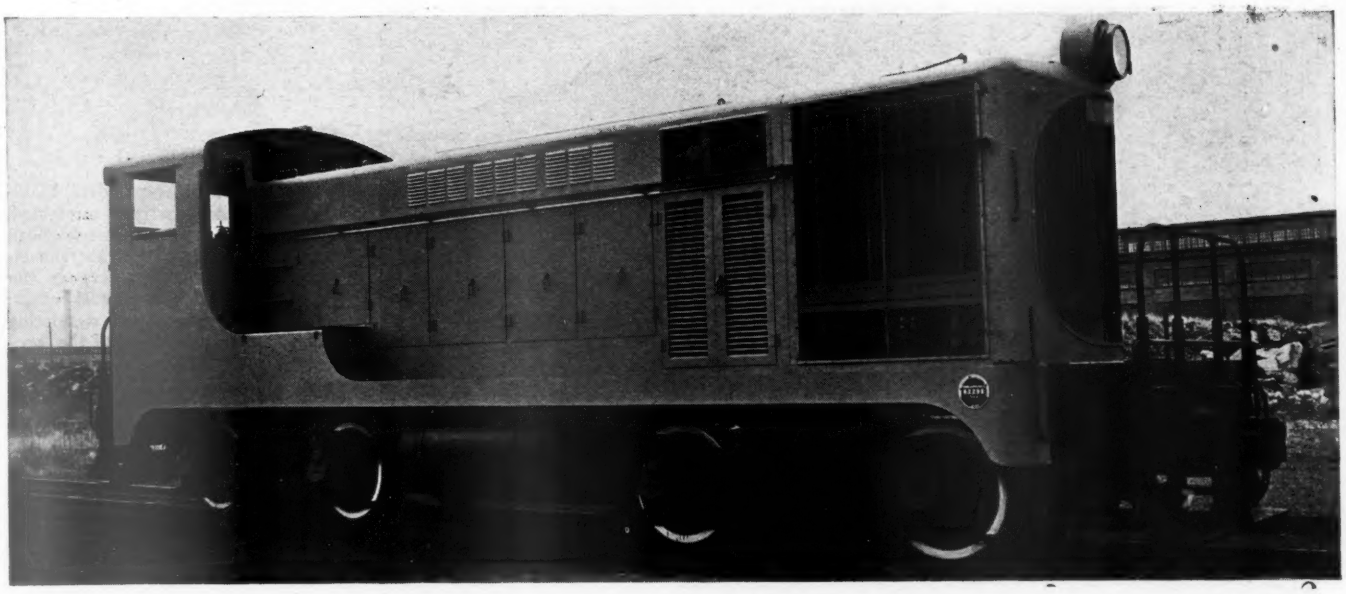
The De La Vergne Model VO Diesel engines operate on the four-cycle principle, using solid injection. The six-cylinder engine is rated at 660 hp. and the eight-cylinder engine 1,000 hp. at 625 r.p.m. Both engines have 12¾ in. by 15½ in. cylinders. Accessibility is obtained by mounting the governor, lubricating-oil pump, fuel-transfer and injection pumps, fuel lines, spray valves and camshaft externally; working parts are enclosed.

A feature of the De La Vergne engine is the spherical water-cooled combustion chamber which is cast integrally with the cylinder head, and connected with the cylinder combustion space by a throat. During the compression stroke the air is forced into the chamber, entering tangentially at high velocity, thereby producing great turbulence. Fuel oil is sprayed into the combustion chamber through a multiple-orifice spray nozzle which has comparatively large openings. This feature, together with the cooling of the nozzle, minimizes clogging. The De La Vergne combustion system permits the injection of fuel over a longer period, at an unusually low injection pressure, resulting in lower firing pressure rise per degree of crank travel, and low maximum combustion pressures.

Cooperative Specifications of Baldwin 120-Ton, 1,000 hp. and 100-Ton, 660-hp. Diesel-Electric Switching Locomotive

	1,000 hp.	660 hp.
Total b.hp. (single Diesel engine)	1,000	660
Engine, no. of cylinders	8	6
Total weight of locomotive, in working order, approx., lb.	240,000	200,000
Length over couplers, ft.-in.	48- 0	45- 0
Wheelbase of locomotive (total), ft.-in.	33- 6	30- 6
Wheelbase of truck (rigid), ft.-in.	8- 0	8- 0
Height from rail to top of cab (maximum), ft.-in.	14- 6	14- 6
Height from rail to centerline of couplers, in.	34	34
Height from rail to cab floor, ft.-in	7- 3	7- 3
Height from rail to top of engine compartment hoods, ft.-in.	12- 3⅛	12- 3⅛
Width (maximum outside), ft.-in.	10- 0	10- 0
Width of cab, ft.-in.	9-10	9-10
Width of engine compartment hoods (outside), ft.-in.	5- 8¼	5- 8¼
Fuel oil capacity, gals.	700	600
Water capacity, gals.	320	240
Lubricating oil capacity, gals.	64	55
Sand capacity, lb.	3,000	3,000
Driving wheels, diameter, in.	40	40
Journal load, lb.	26,400	21,400
Starting tractive force, 30 per cent adhesion, lb.	72,000	60,000
Starting tractive force, 25 per cent adhesion, lb.	60,000	50,000
Maximum speed, m.p.h.	60	45

The fuel injection system is the Bosch solid-injection



The Baldwin 100-ton switcher, powered by a 660-hp. De La Vergne engine

type with spring-loaded needle-type spray valves and multi-hole nozzles. One Bosch fuel injection pump and one spray valve are provided for each cylinder.

The fuel supply is delivered to the injection pumps by a positive-displacement fuel service pump. This pump delivers fuel under pressure through metal-edge strainers and a cartridge-type filter, in series, to the injection-pump supply header. Fuel oil pressure in the supply header is controlled by a spring-loaded relief valve.

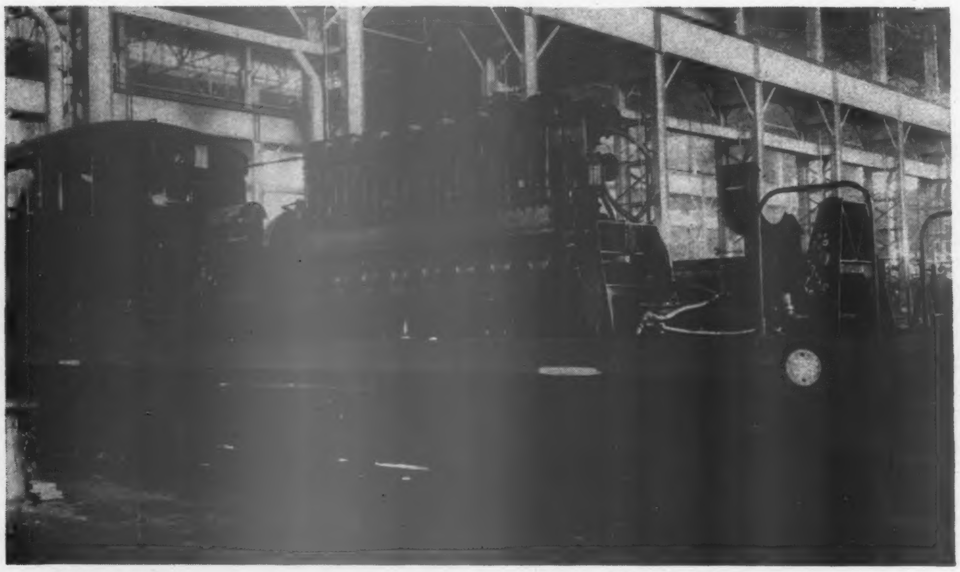
The engine bedplate is made of welded steel with heavy transverse webs to support the main bearings and to maintain proper crankshaft alignment. The main bearing caps are secured to the bedplate by internal throughbolts. The frame is a welded steel structure which forms the cylinder housing and upper part of the crankcase.

The main bearings consist of interchangeable steel shells with centrifugally poured babbitt. All main bearings are adjustable by means of laminated shims. The lower shell may be rolled out with the crankshaft in place. The cylinder liners are of a stress-relieved iron. They are machined all over, and finished internally by honing. Each liner has a flange at the top which seats in the engine frame, and is provided with rubber rings at the lower end to seal the joint between the liner and the frame.

The cylinder heads are stress-relieved iron castings. The turbulence combustion chamber is located in the cyl-

Curve showing current consumption and tractive force in relation to speed for the 1,000 hp. locomotive—The gear ratio is 16 to 59

Tractive force—speed—current consumption curves for the 660 hp. Diesel-electric locomotive with a gear ratio of 16 to 76



One of the eight-cylinder, 1,000-hp. switchers partially assembled. The control panel compartment can be seen ahead of the cab; the air compressor is located between the generator and the control panel; the radiators and air circulating fans at the forward end of the locomotive have not yet been mounted

inder head, offset from the cylinder center, and is surrounded by cooling water. This arrangement permits the placing of the spray valve horizontally at the side of the head, in an accessible position, and provides cooling for the spray valve.

One exhaust and one inlet valve is arranged in the head for each cylinder. They are actuated by rocker arms mounted on brackets attached to the cylinder head. The rocker arms are fitted with rollers to contact the valve stems and are actuated by hollow push rods, socket-mounted in roller-type cam followers. The camshaft is located in a housing on the control side of the engine. The camshaft bearings are removable.

The crankshaft is a solid forging of heat-treated, open-hearth steel, drilled for pressure lubrication.

The connecting rods are made of die-forged alloy steel, heat-treated. The wrist-pin end is fitted with a one-piece bushing of high-grade bearing bronze. The crank-pin bearings have centrifugally cast, babbitt-lined steel-backed shells, fitted with laminated shims. Two heat-treated alloy-steel crank-pin-bearing bolts are used for each rod.

The hollow bored wristpins are of the full floating type, made of alloy steel forgings.

The pistons are made of aluminum alloy. Each piston carries five compression rings and has one lubricating oil control ring above the wristpin and one below.

The Woodward governor is gear-driven from the camshaft. It is of the hydraulic relay type and its speed setting is determined by an electro-pneumatic governor operator.

Lubrication is furnished by a pressure system to the crankshaft, connecting rods, wrist-pin bearings, cylinder liners, camshaft bearings and valve rocker arms. An automatic lubricating-oil safety stop is applied to shut off the fuel supply to the injection pumps in the event that the lubricating-oil pressure should drop below a safe minimum.

Electrical Transmission

The engine is directly coupled to a Westinghouse generator which supplies power to four axle-hung force-ventilated series-wound Westinghouse traction motors. The generator for the 660-hp. locomotive has a continuous rating of 1,000 amp., while that for the 1,000-hp. locomotive is rated 1,200 amp. Exciters are mounted on the generator shafts.

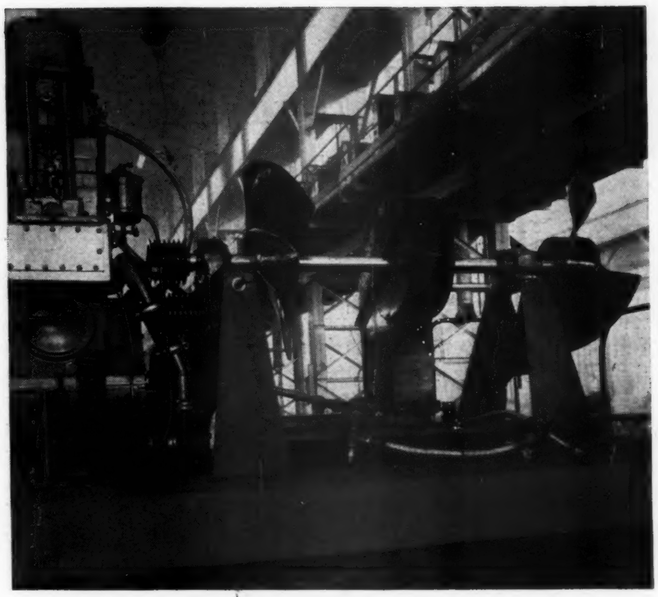
Baldwin 120-ton, 1,000-hp. switcher built for the Santa Fe

The continuous ratings of the motors are 470 and 560 amp. when blown with 1,100 and 1,200 cu. ft. of air per minute, respectively. Two blowers supply cooling air for the traction motors. They are mounted at opposite ends of the engine-generator-compressor unit and are driven by V belts taking power from the main shaft.

Electro-pneumatic control is used to operate two motors in series, two in parallel, full field, and two motors in series and two in parallel, shunted field. The engine governor is electro-pneumatically controlled from the master controller. Excitation of the main generator is automatically controlled to utilize the full output of the engine over its operating range, and avoid overloading.

An auxiliary generator furnishes power for charging the battery, and for the operation of control circuits and lights. It is mounted on top of the exciter frame and is driven from the generator shaft by grooved pulleys and V-belts.

The storage battery on the stock locomotives consists of 56 cells of Philco 17-XV on the 1,000-hp. unit and 56 cells of Philco 21-KRT on the 660-hp. These are rated at 288 and 240 amp.-hr. respectively. The five Santa Fe 1,000 hp. locomotives were equipped with Exide 56-cell, 288 amp.-hr. batteries. The batteries are used for engine starting and standby lighting and are charged from the auxiliary generator.



The air circulating fans, before the installation of the radiators

Other Construction Features

The underframes for both the 660- and 1,000-hp. locomotives are cast steel, manufactured by the General Steel Castings Corporation. The cab is located at one end of the locomotive. Directly ahead of the cab, under the hood, is the electrical control panel, then the compressor, the generator and engine unit and, at the other end of the locomotive, the blower, fans and radiators.

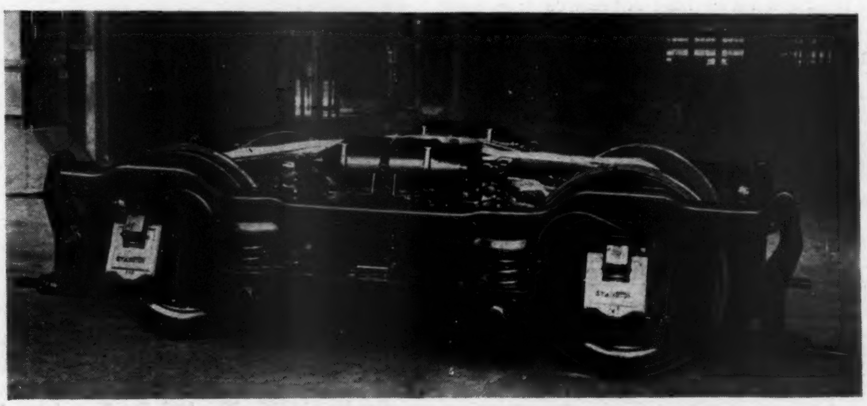
The cab is built of steel plates, with wood floor, and lined with Masonite insulating board. The cab doors are steel, and the windows extruded aluminum shapes. All windows are glazed with Saftee glass. The side windows are of the sliding type and have locating latches. The operator's control station is at the right-hand side of the cab. The operator has direct clear vision over and alongside the hood. Three pneumatic window wipers are part of the cab equipment.

The engine radiator compartment is located at the front end of the locomotive. Two fans, driven by V-belt from the engine shaft, draw air into a chamber and blow it out through the two radiators located at either side. On the same shaft is the blower for the traction motors on the front truck. Adjustable radiator shutters permit engine operation at correct temperatures.

The fuel tank for the 1,000-hp. locomotive has a total capacity of 700 gal., that of the 660-hp. locomotive, 600 gal. Both are made of welded steel plates. They are mounted crosswise under the main frames between the trucks. The tank has a filler cap to permit filling from either side of the locomotive, and four sight gages, two located on each side. The electrically-driven gear-type fuel-oil transfer pump takes its suction directly from the fuel tank.

The trucks for stock locomotives are the General Steel Castings four-wheel type. The frames are cast steel with pedestal jaws and center plates cast integral. Double helical and semi-elliptic springs support the truck frame. The semi-elliptic springs are swung on forged links held by two equalizers on each side of the truck. The equalizer ends rest in seats on the journal boxes. Hollow transoms conduct the air from the blowers to the traction motors. The Santa Fe 1,000 hp. locomotive shown in one illustration is equipped with trucks of Baldwin design. The motors are supported between the driving axles and the spring motor-nose suspension on the truck transoms. The axle journals are 6½ in. by 12 in. The axle diameter through the motor bearing is 8¼ in. diameter. The rolled-steel wheels are 40 in. in diameter. The journal boxes are cast steel with forged-steel wedges and A.A.R. railway type crown brasses. A thrust bearing is provided in each box for absorbing the lateral thrust of the axle. The side bearings are hardened steel

(Continued on page 64)



Four-wheel truck of the type used on the stock locomotives

Comparative Service Tests of

USS Cor-Ten Vs. Copper Steel

Report of service life of hopper cars in coke-carrying service built with copper steel in one end and USS Man-Ten or USS Cor-Ten in the other end

When USS Cor-Ten was introduced for railroad car construction, the results of atmospheric corrosion tests were available which showed the resistance of this material to corrosion in various types of atmospheres to be four to six times that of carbon car steel, or two to three times as much as copper steel. This superior corrosion resistance was due partly to the fact that the rust which formed on USS Cor-Ten was harder than that on carbon steel or copper steel and that it adhered more tenaciously to the base metal. In view of the differences between the conditions in an atmospheric test and those existing in open-top freight cars, the question arose whether the abrasive action of the lading, especially in hopper cars, would remove the coating from Cor-Ten and cause rusting to proceed in cars more nearly at the same rate as with copper steel.

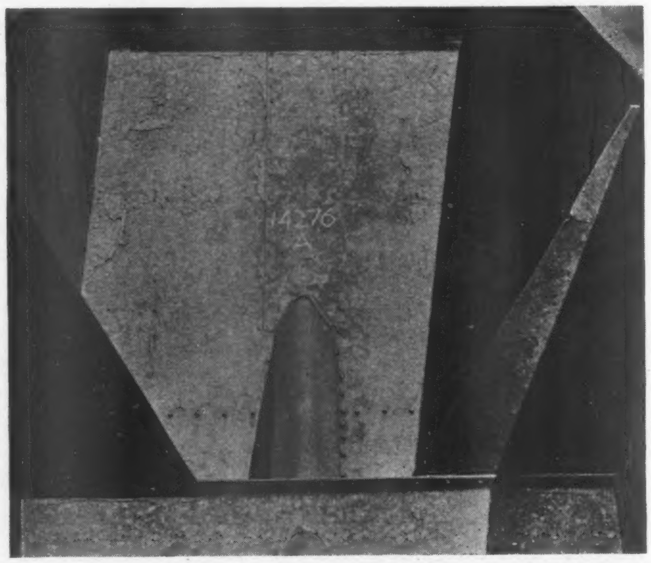
As this question seemed to have an important bearing on the advantage to be derived from the use of USS Cor-Ten in open-top cars, the rate of rusting of cars resulting from abrasion and the action of corrosive leachings from coal was studied by the Corrosion Research Laboratory of the Carnegie-Illinois Steel Corporation. The results of this investigation indicated that the corrosion of steel cars was caused principally by exposure to atmospheric conditions and that longer service life would be obtained by using a steel such as Cor-Ten which is much more resistant to severe industrial atmospheres than plain or copper steel.*

In order to obtain results from actual service as quickly as possible some 50-ton hopper cars owned by the Carnegie-Illinois Steel Corporation were repaired in 1934, using one type of steel in the *A* end of the car and another type in the *B* end. The materials applied for the test included Cor-Ten, an intermediate manganese steel, and copper steel. Where copper steel was used, the floors, hoppers, longitudinal hoods, and cross hoods were all 1/4 in. thick, and the sides and ends 3/16 in. thick. The high-tensile steels were applied to some cars in these same thicknesses and in other cars the bottoms were made 3/16 in. thick and the sides and ends 1/8 in.

All of the test cars were placed in service in November, 1934. Some have been in coal service, but the majority have been handling furnace coke between the Clairton By-Product Coke Works and Carrie Furnaces, making frequent trips of 6.6 miles loaded with coke and returning empty. The cars have had intensive use and carried an average of 567 loads up to July 1, 1939, or 10.2 loads per month.

Since the end of the first year's service the cars have been inspected thoroughly at intervals of about six months. As the principal object of the test was to determine the ultimate life of the material, these inspections have consisted of noting the condition of the bodies and especially any perforations in the sheets, and the extent to which the sheets were corroded and worn away. When the test is completed, the loss of weight of each part will be determined.

The condition of the cars now gives definite indications of the relative life to be expected from the materials tested in coke cars and also in other service. Perforations have developed in a large majority of the longi-



Car No. 14276—The copper-steel sheets in the A end were found perforated after 30 months' service

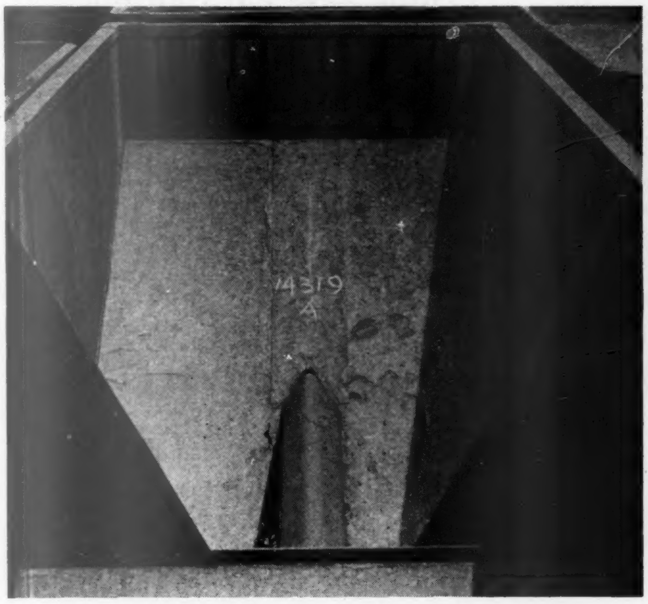
Car No. 14276—The USS Cor-Ten sheets in the B end showed no defects after 57 months' service

* See "Corrosion of Steel Cars," by G. N. Schramm, E. S. Taylerson and C. P. Larrabee, Railway Age, November 28, 1936, page 780.

tudinal hoods, hopper sheets, and floors. In some cars such failures have occurred in all three of these parts. The sides and ends have shown no failures up to this time.

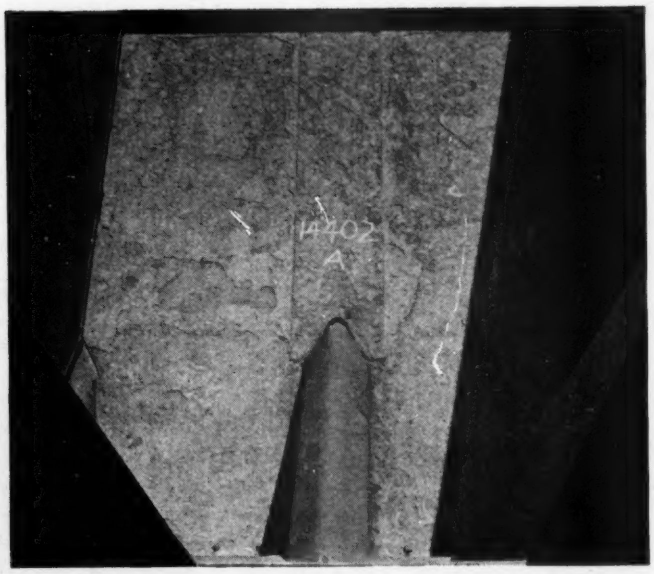
Of the five cars with ¼-in. copper-steel floors, three were found perforated after 30 months' service and two after 42 months' service. The ¼-in. floors of intermediate manganese steel were perforated in the same period, but the general condition of the sheets was better than the copper steel. Three cars with ¼-in. Cor-Ten floors show no defects after 57 months, indicating that in coke service Cor-Ten will last twice as long as copper steel of equal thickness.

Of the two cars with $\frac{3}{16}$-in. Cor-Ten floors and hoppers, one showed the first perforation after 42 months and the other was intact after 57 months. On the basis of comparing the relative life of copper and Cor-Ten steels after the first perforation the Cor-Ten steel, in 25 per cent thinner gage than copper steel, still gave 40 per cent longer life.



Car No. 14319—Copper-steel sheets in the A end were found perforated after 42 months' service

Car No. 14319—The USS Cor-Ten sheets in the B end showed no defects after 57 months' service



Car No. 14402—The copper-steel sheets in the A end were found perforated after 30 months' service

Car No. 14402—The USS Man-Ten sheets in the B end were perforated after 30 months' service

The performance of the intermediate manganese steel in this test brings out some significant facts. This steel has approximately the same strength and abrasion resistance as Cor-Ten. In resistance to atmospheric corrosion, intermediate manganese steel is equal to copper steel. Even though the abrasive action of coke on the floors of these cars is severe, the copper steel lasted almost as long as the intermediate manganese steel. The results indicate definitely that the life of the floor sheets in these hopper cars is affected only to a minor degree by variations in strength or abrasion resistance of the steels tested. On the other hand, Cor-Ten, which, in addition to higher strength and abrasive resistance, has superior resistance to atmospheric corrosion, gave increased life over the other steels almost in the same ratio as the increase noted in atmospheric corrosion tests.

Among the diversified conditions to which steels are subjected in car service, these coke hopper-car floors,

(*Continued on page 64*)

Denver and Rio Grande Western

Automobile-Box Cars

These cars have an unusual door arrangement which facilitates their use in either the transportation of automobiles or grain and other box-car commodities

Above: The large door closed and the standard 6-ft. box-car door open, ready for the loading of box-car commodities—Right: The clear opening is 15 ft. 1 in. wide when both doors are open for the loading of automobiles

One hundred automobile-box cars of 50 tons capacity were recently delivered to the Denver and Rio Grande Western by the Pressed Steel Car Company. The cars are of lightweight construction and embody a new door arrangement which utilizes an auxiliary inside door, patented by W. H. Sagstetter, chief mechanical officer, Denver and Rio Grande Western.

The cars have two doors on each side of different widths which enables the door openings to be made 15 ft. wide by 10 ft. ¾ in. high for the loading of automobiles, or with a standard 6-ft. width for the loading of box-car commodities. When the cars are used for the latter purpose, the auxiliary inside door slides over the large automobile door opening, forming a separate wall, and thereby eliminates the installation of any temporary lining. These inside doors are of corrugated-steel construction and are lined with plywood. They fit in a recess in the side wall when not in use in such a manner as not to interfere with the loading of any commodity. The doors have safety catches of the gravity type to hold them automatically in a closed or open position.

Underframes and Trucks

The underframes have the Duryea cushioning arrangement. The center sill consists of two 10-in. open-hearth steel channels with top and bottom cover plates applied with a continuous weld. The side sills are 10-in. high-tensile-steel channels, running from end sill to end sill, with a reinforcement consisting 6-in. bulb angles attached to the side sill under the door opening. The end sills are high-tensile steel angles.

The body bolster is of the built-up type of welded construction having $\frac{3}{16}$-in. high-tensile steel web plates and ⅜-in. top and bottom cover plates. The four cross bearers are also of high-tensile steel of welded construction extending the full width of the car. The diagonal bracing is of ⅛-in. high-tensile steel extending from the corner of the car at the junction of the side and end sills to the junction of the bolsters and the center sill. The body center plates are of high-tensile cast steel conforming to the A. A. R. contour, and each plate is held in position by ¾-in. rivets reinforced by welding. Three-inch Z-bars of high-tensile steel form the floor stringers which are welded to the cross bearers and bolster web.

The cars are equipped with self-aligning trucks of the Truck Association design having a spring plankless double-truss frame. The side frames and the truck bols-

ters, made by the American Steel Foundries, are of high-tensile cast steel. The side frames have the brake-hanger bracket cast integral and the latter provides for the application of the Mobil brake-hanger retainer. The side bearings are of the roller type. All helical springs

Principal Weights and Dimensions of the Denver & Rio Grande Western Automobile-Box Cars Built by the Pressed Steel Car Co.

Length over coupler carrier casting, ft.-in.	42-10¾
Length over end sills, ft.-in.	40- 8⅛
Width over side sills, ft.-in.	9- 9⅝
Length, inside, ft.-in.	40- 6 1/16
Width, inside, ft.-in.	9- 2 1/16
Width, side door opening, clear, automobile car, ft.-in.	15- 1
Width, side door opening clear, box car, ft.-in.	6- 0
Height, inside at side plate, ft.-in.	10- 8⅛
Height, inside at center, ft.-in.	11- 1 3/16
Height, rail to top of running boards, ft.-in.	15- 1
Cubic capacity, cu. ft.	3,961
Light weight, lb.	44,000
Load limit, lb.	124,600
Ratio, pay load to gross load, per cent	73.7
Journal sizes, in.	5½ x 10

were furnished by the American Locomotive Company, Railway Steel Spring Division, but 50 cars were equipped with the Holland Volute spring snubbers and the remaining 50 cars with the American Steel Foundries' snubbers. The trucks have one-wear steel wheels, standard A. A. R. journal bearings, pressed-steel journal-box lids, and Jenkins leather fibre dust guards and wedges. Fifty cars have Creco brake beams and the other 50 cars have Davis brake beams. The brake shoes, with reinforced back, are 2 in. thick, and are held by Lockey brake-shoe keys. Schaefer brake hangers of the integral-forged type and the Grip Nut safety support and safety device, with the bottom rod going through the bolster, were installed.

The design of the superstructure follows the recommended A. A. R. practices. The roofs are the Murphy solid-steel riveted type using No. 16 gage USS Cor-Ten steel sheets and ⅛-in. USS Cor-Ten steel caps with a provision for the application of automobile loading de-

Partial List of Materials and Equipment on the Denver & Rio Grande Western Automobile-Box Cars

Underframe	O. C. Duryea Corp., New York
Roof	Standard Railway Equipment Mfg. Co., Chicago
Running boards	Apex Railway Products Co., Chicago
Outside doors	The Youngstown Steel Door Co., Cleveland, Ohio
Ends	Union Metal Products Co., Chicago
Side and end ladders	The Wine Railway Appliance Co., Toledo, Ohio
Floor clips	Western Railway Equipment Co., St. Louis, Mo.
Leak-proof bolts; grip-holding nuts	Grip Nut Co., Chicago
Coupler	National Malleable and Steel Castings Co., Cleveland, Ohio
Uncoupling rigging	Union Metal Products Co., Chicago
Truck side frames; bolsters	American Steel Foundries, Chicago
Side bearings	A. Stucki Co., Pittsburgh, Pa.
Journal bearings	Magnus Metal Div., National Lead Co., New York
Steel wheels	Carnegie-Illinois Steel Corp., Pittsburgh, Pa.
Helical springs	American Locomotive Co., Railway Steel Spring, Div., New York
Snubbers	(50) Holland Company, Chicago (50) American Steel Foundries, Chicago
Brake hanger retainer	Illinois Railway Equipment Co., Chicago
Brake beams	(50) Chicago Railway Equipment Co., Chicago (50) Davis Brake Beam Co., Johnstown, Pa.
Brake shoe; brake shoe keys (Lockey)	American Brake Shoe & Foundry Co., New York
Brake hangers	Schaefer Equipment Co., Pittsburgh, Pa.
Air brakes, type AB	Westinghouse Air Brake Co., Wilmerding, Pa.
Hand brake	Ajax Hand Brake Co., Chicago
Brake step	Apex Railway Products Co., Chicago
Brake regulator	Royal Railway Improvements Corp., New York
Journal box lids	American Locomotive Co., Railway Steel Spring Div., New York
Dust guards; wedges	Portable Plating & Equipment Co., Chicago
Angle-cock holder; defect card holder; pipe clamps	Railway Devices Co., St. Louis, Mo.
Retainer pipe clamp	Illinois Railway Equipment Co., Chicago
Safety support and device	Grip Nut Co., Chicago
Draft-key lock	Illinois Railway Equipment Co., Chicago
Paint and primer	(50) E. I. du Pont de Nemours & Co., Inc., Wilmington, Del. (50) Wardway Paint Works, Chicago Heights, Ill.
Paint and primer, roofs	The Glidden Co., Cleveland, Ohio

Left: When the car is used for the transportation of automobiles the inner door slides back in a recess in the side wall—Right: The inner door is in position for the loading of grain and other box-car commodities

vices. The Apex steel running boards, made in sections and fitting between the roof seam caps, and steel latitudinal running boards were applied. The outside doors were made by the Youngstown Steel Door Company and are fitted with roller lift fixtures. The cars have Dreadnaught two-piece steel ends with round corners with the top and bottom sections of 1/8-in. and 5/32-in. USS Cor-Ten steel, respectively. The side sheets are of No. 14 gage USS Cor-Ten which are riveted to the posts as well as to the side plates and side sills.

The flooring consists of fir with a 5 1/4-in. face having calking cement on the ends of the boards with the top of the floor sanded. The floor boards are held in position by floor clips, leak-proof bolts, and grip holding nuts. The side lining is of 25/32-in. fir, tongue and groove running horizontally, with the lining in the pocket behind the inside door of 3/4-in. plywood. The end lining is also of 25/32-in. fir running vertically.

The cars are equipped with Type-E rotary couplers of the bottom-operating type made of Grade B cast steel by the National Malleable and Steel Castings Company, draft keys of open-hearth steel, and the Imperial rotary uncoupling rigging. The brake equipment is comprised of the Westinghouse Type-AB air brake and the Ajax vertical-wheel hand brakes. The side and end ladders were furnished by the Wine Railway Appliance Co.

A Study of The Locomotive Boiler

(*Continued from page 51*)

pass through this furnace in 0.1 second and this is one of the reasons for the great loss of carbon at high-capacity operation. As a comparison, in stationary practice the velocity of the gases just before entering the tube bank is about 10 m.p.h. at a maximum. A study of the designs of many recent boilers indicates that the width of the firebox, as well as the depth, could have been increased to give the advantage of a larger combustion volume under the firebrick arch, as well as a larger gas area between the top of the arch and the crown sheet.

A few modern boilers have been built with a firebox ring as low as 53 in. from the rail which, with a horizontal grate, would give a good depth of the firebox at the rear. On many locomotives with the grate sloping up toward the back end, the distance from the rail to the firebox ring is 72 in., or more, and thus there is a loss of 18 in. in the depth of the firebox. The author has not been able to get a satisfactory reason why sloping grates are still continued. Many boilers have been built with horizontal grates and have given satisfactory service.

(*To be concluded*)

USS Cor-Ten Vs. Copper Steel

(*Continued from page 61*)

with their relatively short life, represent one extreme of intensive service in which the effect of atmospheric corrosion in relation to total deterioration is at a minimum and abrasion is at the maximum. At the other extreme are those parts of cars which receive no abrasion and last as long as they can resist atmospheric corrosion. Tests in which steels have been subjected to exposure in rural, industrial, and sea-coast atmospheres without abrasion have shown that, under these conditions, Cor-Ten has two to three times the life of copper steel. It is noteworthy that in the hopper-car test described above, where conditions are so different, Cor-Ten again shows almost the same degree of superiority. Because the two types of tests representing opposite extremes of service conditions give such similar results, it is believed that it is conservative to estimate that in any application in freight-car construction Cor-Ten will develop approximately twice the life that would be obtained from the same thickness of copper steel.

Baldwin Diesel-Electric Switchers

(*Continued from page 59*)

plates with shims for adjustment. Alemite grease lubrication is applied to the truck center pin and oil lubrication is used for the pedestal gibs and journal thrust bearings. The trucks are equipped with clasp brakes.

The equipment ahead of the cab is covered by a 68 1/4-in. hood built of sheet steel on a steel frame and designed with doors on the sides. The doors are equipped with latches and locks, are hinged and when open give access, by means of the 25-in.-wide roughened plate platform inside of the hood, to all equipment located therein. Ventilation is effected through screened and baffled side openings. The hood is equipped with top inspection hatches and with lifting bales for removing it as a complete unit. Permanent inspection-light fixtures are located inside the hood.

The air-brake equipment is Westinghouse Schedule 14-EL. Both straight and automatic air brakes operate on all wheels. There are two air reservoirs with a capacity of 60,000 cu. in. The locomotives are equipped with one Ajax hand brake with a hand-wheel located in the cab and connected to the brake rigging of one truck to hold the locomotive while out of service.

Compressed air is furnished by a Gardner-Denver three-cylinder two-stage air-cooled compressor having a displacement of 56 cu. ft. of free air per minute at an engine idling speed of 250 r.p.m., and 140 cu. ft. at full engine speed of 625 r.p.m. The air compressor is driven by a Thomas coupling from the generator shaft.

Accessories used on these locomotives include the following: Miner A-22XB friction draft gear; National Type E swivel shank, top-operating couplers; 12-in. bell with internal, quick acting pneumatic ringer; Pneuphonic horn; Sunbeam headlights and Leach Type D pneumatic sanders.

* * *

Photo by Granville Thomas

On the Pennsylvania-Reading Seashore Lines at Millville, N. J.

EDITORIALS

Locomotive Availability

In motive-power economics the outstanding advantage of the steam locomotive is its relatively low first cost. In road service where the locomotive unit must have approximately the same horsepower capacity, whether steam or Diesel-electric, the first cost of the Diesel unit will range from two and one half to three times that of the steam locomotive. One of the strongest claims for the Diesel-electric locomotive, however, is its high degree of availability as compared with that of the steam locomotive, which is an offsetting factor.

There is no opportunity to compare the degree of intensity with which steam and Diesel-electric locomotives are used in freight service since the Diesel-electric locomotive has not yet accumulated any records in this service. In passenger service, however, data enough are available to give some indication of the possibilities which the Diesel-electric locomotive offers in this respect.

During 1939 an ownership of 7,320 steam passenger locomotives averaged 45,000 miles per year per locomotive. This is an average of 3,750 miles per locomotive per month, or 125 miles per locomotive day on the basis of ownership. The report of the Committee on the Utilization of Locomotives presented at the meeting of the Railway Fuel and Traveling Engineers' Association last October quoted an average daily mileage per *active* passenger locomotive during the first six months of 1939 of 183.5. The best average mileage for a single railroad was 269.7 per active locomotive day and the lowest 144.7.

Compare this with the performance of Diesel-electric passenger locomotives during 1939. The 58 locomotives in passenger service (as of June 30, 1939) averaged 178,600 miles per year. This is equivalent to 14,900 miles per month and 497 miles per day. While this number of locomotives probably does not accurately represent the total number of locomotive days during the year because of additions during the last half of the year, the error is not believed to be large. In each of 1937 and 1938, with a much closer average of the number of locomotive days and a much smaller total locomotive mileage, Diesel-electric locomotives averaged well over 400 miles per day.

In interpreting these figures account must be taken of the fact that they represent the aggregate performance of a number of individual units assigned to runs on which the conditions are ideal for a high degree of utilization, whereas the steam-locomotive performance represents complete power assignments in which are included spare power and runs on which high mileages are impossible. But even when the active steam locomotive mileage is compared with the mileage of the Diesel-electric locomotives owned, the mileage performance of the latter is outstanding.

This situation suggests that one of the most important factors in the future development of the steam locomotive must be the improvement of its capacity for more intensive utilization. Indeed, it is a question if the steam locomotive is not, in this respect, a victim of its own history. Until 20 years ago scarcely anyone thought it practicable to operate a steam locomotive over more than a single crew district. Even before long runs were being taken into account in locomotive design, however, the steam locomotive demonstrated its capacity for runs as long as traffic and road characteristics permitted. Some of the more recently built locomotives have demonstrated their ability to remain out of the shop for classified repairs for well over 200,000 miles. It does not seem beyond the possibilities of the future to remove some of the other limitations on continuity of service which still have to be contended with.

Steam and Diesel Locomotive Comparisons

In a comment on the comparison of the performance and operating costs of Diesel-electric locomotives and steam locomotives in these columns last month, comparisons on three points were discussed. These were the question of fuel cost, the question of the horsepower-weight ratio, and the cost of maintenance. No mention was made in this discussion of the relative effect of first cost when the comparison is between the Diesel-electric locomotive and the new steam locomotive for the same service.

In considering this aspect of the relative economic value of the Diesel-electric and steam motive power there are really two comparisons which have to be considered in every specific instance. The first is that between the old motive power by which the service is now being performed and the proposed new steam motive power. The second is between the proposed

Diesel-electric motive power and the existing steam motive power. The effect of the value of the existing steam motive power on such first-cost comparisons will depend on its bookkeeping status. If all of its first cost, except the usual allowance for recoverable scrap value, has been recovered through depreciation charges, the ultimate effect on the accounting of the retirement of the existing power and its replacement with new will be in the property account and will be the difference between the first cost of the new power to be purchased and the original cost of the existing power. If, however, the existing power has been subject to inadequate depreciation charges, a retirement charge will have to be made in operating expenses, as a part of the transactions involved in replacing the old power with new. This will operate against the new power only during the year in which it is made. In setting up the comparison, however, the undepreciated portion of the first cost of existing equipment is sometimes added to the first cost of the new equipment for the determination of the carrying charges which must be earned by the new locomotives before they can be considered as moneymakers. In any case, the ultimate comparison between the Diesel-electric locomotives and the new steam locomotives would involve the balancing of the difference in their operating costs against the carrying charges on the difference in their first costs.

In the case of switching locomotives, the difference in first cost of new motive power of the two types would be at its lowest and operating advantages probably the most favorable to the Diesel-electric locomotive. In the case of road locomotives, with the horsepower capacity of the two types about equal, the difference in first cost will be greater and operating costs less favorable to the Diesel-electric.

In either case, however, the carrying-charge handicap of the Diesel-electric is less when compared with a new steam locomotive than when compared with an old steam locomotive, the first cost of which has been partially recovered through depreciation.

The Designer Must Consider Maintenance

One of the points stressed by C. A. Brandt, chief engineer of the Superheater Company, in his paper on the locomotive boiler presented at the 1939 annual meeting of the American Society of Mechanical Engineers, was the importance of obtaining as large a gas area as possible through the boiler. The reasons for this are discussed in Mr. Brandt's paper, the first part of which appears elsewhere in this issue. There are also logical reasons why a large gas area may create conditions which are undesirable from the standpoint of maintenance.

In presenting the committee report on What Is the Real Cause for Flues Cracking Longitudinally Through the Beads at the 1939 annual meeting of the Master Boiler Makers' Association, Chairman E. E. Owens stated "It is felt by many, especially the boiler maintainers, that the engineer designing the boilers is too prone to obtain the greatest amount of flue gas area, possibly by crowding in flues to the extent that the water space between the tubes and flues is too close and with a small accumulation of scale the circulation of water is restricted to a point where the heat is not absorbed fast enough, allowing the beads and flue ends to become overheated." It was the committee's suggestion that consideration be given to the spacing of flues with as liberal a bridge as possible and also that the tubes and flues should not be staggered to the extent that washing out between the rows could not be readily accomplished.

No matter how well designed a locomotive boiler may be from the standpoint of ideal proportions, it cannot be safely and efficiently operated without current inspections, tests, and repairs. It is important, therefore, that it be designed to facilitate its maintenance, and thus decrease the time required to perform these necessary operations. The designer who ignores this fact will not only increase the maintenance costs but will also reduce the locomotive's availability, an important competitive factor today.

Boiler Explosions

In the report of the Chief Inspector of the Bureau of Locomotive Inspection for the fiscal year ended June 30, 1939, which is reviewed in this issue, are reported six locomotive boiler explosions caused by crown-sheet failures due to low water. In only two of these were contributory causes or defects found. This is not out of line with the records of the immediately preceding years when, in 1938, five such accidents were reported and, in 1937, eight reported, in three of which contributory causes or defects were found. During the past fiscal year 12 were killed and 11 injured in these accidents; in 1938, 5 were killed and 3 injured; in 1937, 13 were killed and 8 injured. The past year, therefore, shows no significant change either in the number of accidents or in the number of resulting casualties.

Considered together, the three years might imply that an irreducible minimum of such accidents in which no contributing causes or defects are found has been reached at about four or five per year. But accidents of this kind which have occurred since the close of the last fiscal year, on which reports have been issued by the Bureau, leave one with the uneasy suspicion that such a minimum is going to be exceedingly hard to maintain. Reports have been issued on four accidents resulting from crown-sheet failures due to low water

in the investigation of which no contributory causes or defects were found, all of which occurred in a period of about three and one half months since September 26 last. These accidents were the causes of seven deaths and ten injuries.

Accidents of this kind are an exceedingly old story and, considering the matter in a statistical sense, a minimum of four or five per year in operations involving over 40,000 locomotives scattered over an area as wide as the United States may be considered highly satisfactory. Considering the amount of human suffering which is inevitably caused by these accidents, however, the continuance of even one such accident per year must be a cause of serious concern, not in the least mitigated by the fact that the seat of the trouble in many of them lies with the men who suffered most. An attitude of callousness and indifference either on the locomotive or in the shop and engine terminals is the source of so much of the difficulty that a continuous fight must be waged against it.

Journal-Box Temperatures

In a paper before the annual meeting of the American Society of Mechanical Engineers, which is abstracted in this issue, E. S. Pearce presents interesting test data showing the distribution of the heat generated by journal friction, by means of temperature readings taken at 53 locations within the journal box. In general, it may be said that the temperatures accurately measured on the test plant show what would be expected in a qualitative sense from the very character of the journal box and its contents—the journal, the brass, the wedge and the packing.

An interesting point which the tests bring out, however, is the effective part which the movement of the air about the journal box plays in the dissipation of the heat generated from normal operation of the journals in their bearings. In a comparison of the operation of a waste-packed box in air movement equivalent to the speed in miles per hour at which the journal was operating with its operation in still air, temperatures in the moving air 30 to 40 deg. lower were measured in the walls of the journal box. Nearly as great temperature differences were observed in the journal-box packing and the temperatures at the most critical points of the bearing surfaces are about 20 deg. lower at speeds of 40 and 60 miles an hour; even at 20 miles an hour the difference is as high as 10 deg.

What is of greater importance, however, is the clear demonstration that the successful performance of waste-packed journal boxes depends upon careful adherence to those fundamentals which control the generation of heat at the source: clean and properly saturated journal-box packing, skillfully applied in the box.

New Books

A. S. T. M. SPECIFICATIONS FOR PIPE AND PIPING MATERIALS. *Published by the American Society for Testing Materials, 260 South Broad Street, Philadelphia, Pa. 143-page binder with heavy paperboard cover. Price* $1.25.

These specifications for pipe and piping materials for high-temperature and high-pressure services cover various types of carbon and alloy-steel pipe and boiler tubes, etc., including specifications for classification and dimensions of wrought-iron and wrought-steel pipe; carbon and alloy-steel castings for valves, flanges, and fittings; forged or rolled alloy-steel pipe flanges; alloy-steel bolting materials, and carbon and alloy-steel nuts.

WROUGHT IRON, ITS MANUFACTURE, CHARACTERISTICS AND APPLICATION. *By James Aston and Edward B. Story. Published by the A. M. Byers Company, Pittsburgh, Pa. 100 pages, 6¼ in by 9¼ in. Price* $1.

Several new chapters have been added to the second edition of this book, written to serve as a source of up-to-date information on wrought iron for all who are interested in problems of material selection as well as for students in colleges and universities who may some day become responsible for engineering specifications. The book now contains twelve chapters on wrought iron; its characteristics; early and present-day manufacture; the introduction of other ferrous metals; quality standards; specifications and durability testing; forging and bending; welding, etc.

PROCEEDINGS OF THE 1939 ANNUAL MEETING OF THE MASTER BOILER MAKERS' ASSOCIATION. *Albert F. Stiglmeier, Secretary, 29 Parkwood Street, Albany, N. Y. 263 pages. Price, $3.*

This book contains the official proceedings of the twenty-sixth annual meeting of the Master Boiler Makers' Association held at the Hotel Sherman, Chicago, October 17, 18 and 19, 1939. Included in the proceedings are the addresses given by railroad and government officers, three lectures, and eight technical reports. The lectures were delivered on the subjects of embrittlement in the locomotive boiler, circulation of water in the boiler, and the action of steam and water inside the boiler. The technical reports are given in detail and contain information on the training of boilermaker apprentices, welding and alloy steel in locomotive tender construction, circulation of water in the boiler, chemical feedwater treatment, methods for locating the height of crown sheet and water-level indicating devices, the cracking of flues and tubes through the beads, the inspection and cleaning of air reservoirs, and methods for renewing fireboxes. The discussions of these reports by members are also included.

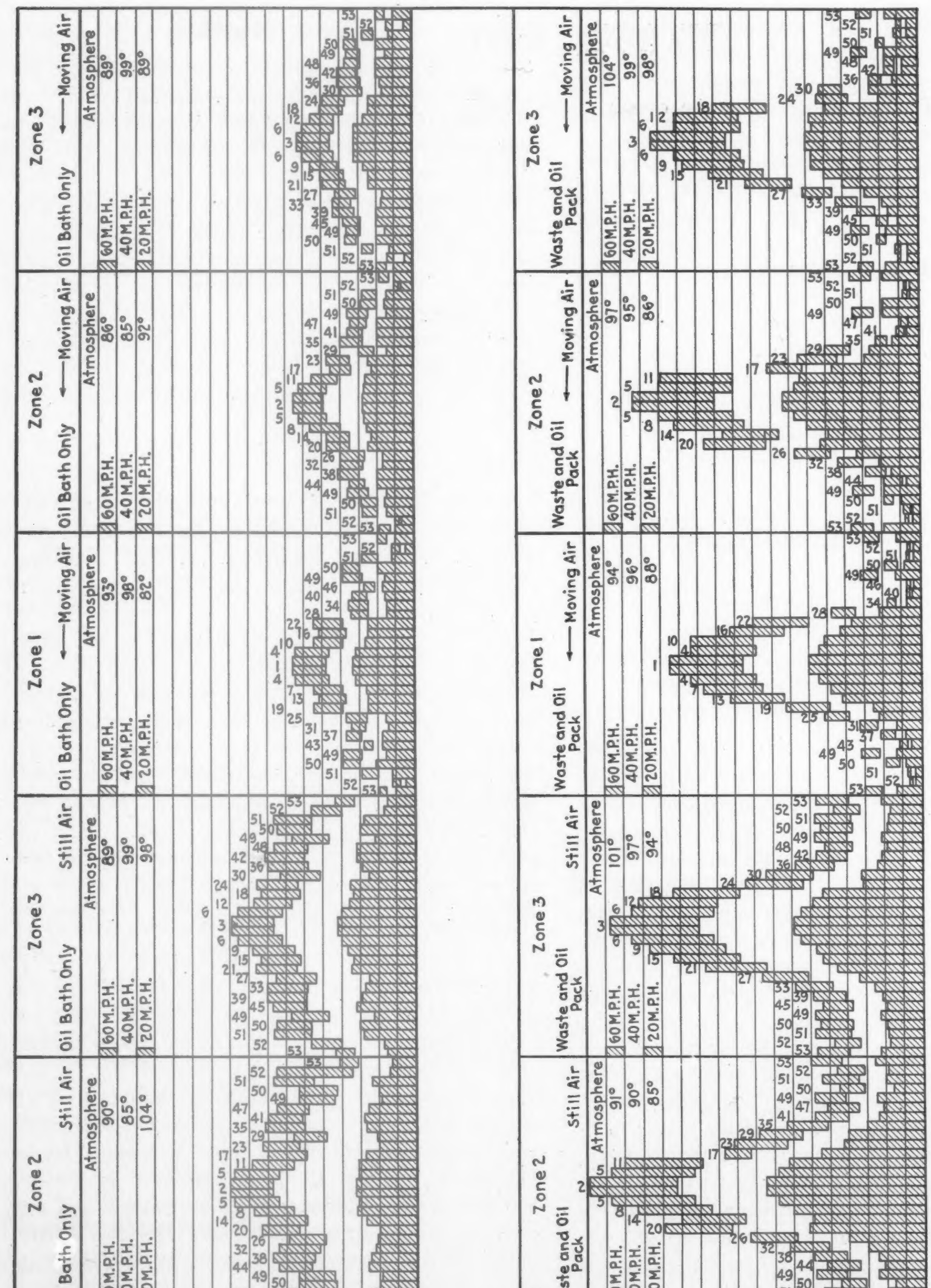


Fig. 1—Temperature distribution in a standard railway journal-box assembly

With the Car Foremen and Inspectors

Distribution and Sources of

Journal-Box Temperature*

By E. S. Pearce†

THE purpose of this paper is to set forth certain data illustrative of the distribution of heat generated in the conventional railway journal-box assembly. These data have been collected collaterally with an extended investigation of heat causes or sources and their control in journal-bearing operation on railroad rolling stock. In a sense, this is an uninterrupted continuation of research and development work described in a previous paper by the author.‡

The same plant and equipment, as described in that paper, were used for these tests, the amplification of equipment being the application of some 53 thermocouples to a 5½- × 10-in. journal box and contained parts. Fig. 2 shows diagrammatically the exact location of all the thermocouples and their numerical designation. Thermocouples Nos. 7, 8, 9, 10, 11, and 12, Fig. 2, are located in the bearing surface to obtain oil-film temperatures.

The preparation for these tests consisted of packing the journal box in the conventional manner with a good grade of wool packing, saturated with oil in the ratio of 3¼ lb. of oil per lb. of waste. The car oil used was of a standard brand, having a viscosity of 45 sec. Saybolt at 210 deg. F. The combined quantity of waste and oil weighed 8 lb., consisting of 1⅞ lb. of waste and 6⅛ lb. of oil. Conventional standard A.A.R. 5½- × 10-in. journal bearings, broached to insure a fixed width of crown, were used. All tests were conducted under a fixed total load of 20,000 lb., giving a unit loading of 890 lb. per sq. in. of projected actual bearing area.

Tests were run at operating speeds of 20, 40, and 60 m.p.h., equivalent to 268, 536, and 804 ft. per min. journal surface speed, with 186, 372, and 558 r.p.m., respectively. A continuous run of seven hours was made at each respective speed, at which time all temperatures recorded were at their stabilized maximum. These temperatures are shown in Fig. 1. Atmospheric temperatures were maintained within practical limits at approximately those of high summer temperatures. Tests were duplicated under conditions of still air and

(*Continued on page 73*)

* An abstract of a paper presented on December 6, 1939, at the annual meeting of the American Society of Mechanical Engineers at Philadelphia, Pa.

† President, Railway Service and Supply Corporation.

‡ "Locomotive and Car Journal Lubrication," by E. S. Pearce, Trans. A. S. M. E., vol. 58, 1936, pp. 37-45.

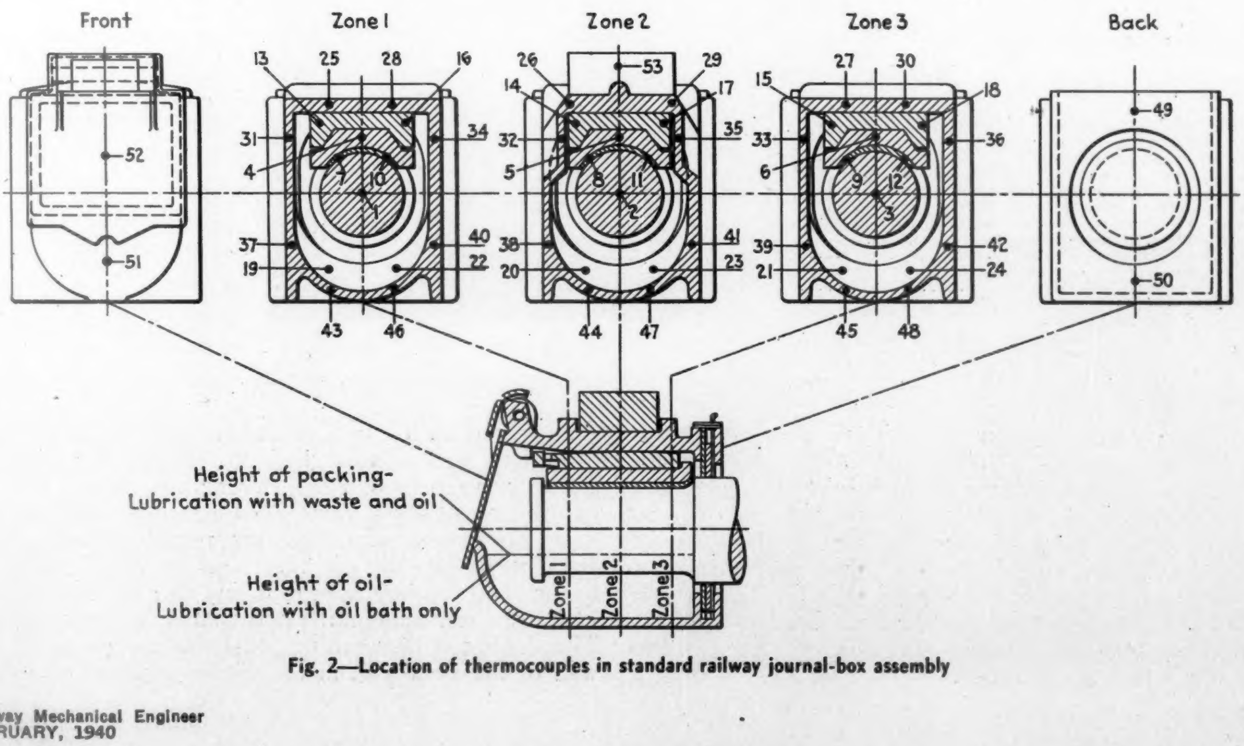


Fig. 2—Location of thermocouples in standard railway journal-box assembly

The inclined runway as seen from the loading end

Improvements At the Readville Shop Facilitate

Demounting Wheels

IN the March, 1936, issue of the *Railway Mechanical Engineer,* an article was published describing the general arrangement and system of operation of the New York, New Haven & Hartford system wheel shop located at Readville, Mass. At the time that article was prepared it was recognized by those in charge of the shop that, while the output was generally satisfactory, there were obstacles in the way of necessary increases in output that would provide the reserve shop capacity quite often needed under unusual conditions. From the standpoint of wheel boring, turning and grinding; the machining of axles and the mounting of finished wheels and axles the shop facilities were entirely adequate. So also was the general arrangement for the handling of wheels and axles within the shop. The opportunity for the greatest improvement appeared to be in the methods used in the demounting of wheels as they came to the shop. It is with the solution of this demounting problem that this article is concerned.

The Change in Facilities

Reference to the shop layout drawing, which appeared on page 113 of the March, 1936, issue, will show that the demounting operations were carried on, under the old arrangement, with one single-end press. The wheels were inspected and checked in the incoming cars just outside the north side of the shop and were delivered, by the two-ton monorail crane, to the floor inside the shop adjacent to the demounting press. After the wheels were placed on the floor in the shop, all of the handling operations were manual, both to the press and after the wheels were demounted. This manual labor, with the one single-end press, constituted a serious handicap.

In considering plans for the re-arrangement of the shop, these two factors were kept in mind. Reference to the shop drawing accompanying this article will show how the wheel demounting department has been changed. The single-end demounting press was moved to the west wall of the shop and, in the corner adjacent to this lo-

Readville Wheel Shop Production—Six Months, 1939

	1	2	3	4	5	6
Cast-iron wheels bored, (no. of wheels)	1,186	576	1,322	1,196	1,086	992
Steel wheels bored, (no. of wheels)	312	434	328	253	248	282
Axles turned and burnished	822	559	977	771	694	626
Treads ground on mounted cast-iron wheels, pairs	22	21	8	15	8	10
Treads ground on mounted steel wheels, pairs	18	17	6	10	14	23
Treads turned on mounted steel wheels, pairs	181	242	282	199	199	194
Journals turned and burnished, mounted wheels, pairs	232	194	231	203	281	210
Cast-iron wheels demounted, pairs	870	555	1,314	1,290	1,950	1,877
Steel wheels demounted, pairs	156	129	105	150	159	154
Cast-iron wheels mounted, pairs	592	283	650	569	540	493
Steel wheels mounted, pairs	116	195	151	94	86	109
Roller bearing wheels serviced, pairs	30	21	81	66	43	37
Eight-hour days worked	21	19	23	20	22	22
Average number of men in wheel shop during month	20	18	20	20	18	18
Average number of pairs of wheels turned out of shop per day	51	39	53	50	47	47

cation, a Chambersburg double-end demounting press was installed. The necessity for manual labor in the handling of wheel sets to this mounting press and the disposal of wheels and axles from the press was largely eliminated by the installation of three devices—a 55-ft. inclined runway feeding wheel sets to the press; two wheel chutes, or runways, for the demounted wheels destined for the scrap car, and an axle chute from the press to the floor. The demounted wheels routed to points within the shop for subsequent use are handled by the overhead monorail.

The location of this overhead monorail was changed, in the demounting section, to accommodate the new location of the presses.

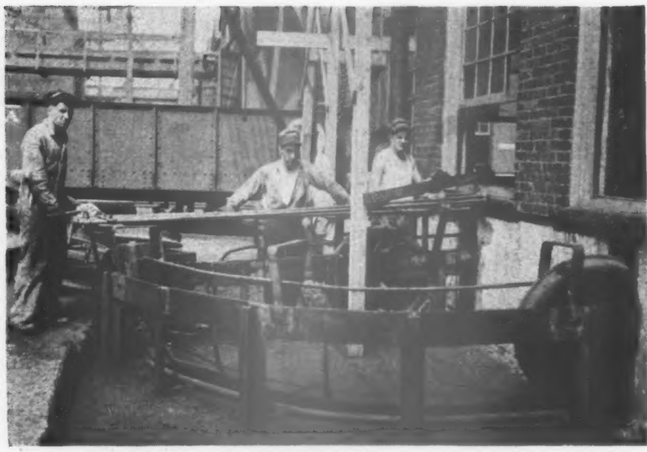
The Inclined Runway

The inclined runway, which now feeds the wheel sets to the double-end press, has a total length of 55 ft. and a slope of 10 in. in that distance. It consists of an elevated standard-gage track with a walkway of planks between the rails. The construction is clearly shown in the illustrations. The entire runway has a capacity of 17 pairs of mounted wheels. This is the number that can be loaded on the special wheel cars now used on the New Haven between outlying points and the central wheel shop at Readville. In operation, the 17 pairs of wheels are unloaded from the wheel car outside the north side of the shop by the two-ton Shepard monorail crane. This crane system serves the entire wheel shop, both inside and out. The wheels are picked up, a pair at a time, and brought into the shop and deposited at the high end of the runway. Each pair is then rolled down, under control, to a pneumatically-operated wheel stop device which controls the movement of the wheels to the press one pair at a time. One of the illustrations shows the arrangement of this device which consists of an air cylinder and two pairs of cam-like wheel stops. The device is controlled by a two-way valve which, in one position, throws the upper stops across the runway rails. When the valve is in the other position, the wheel sets can run down the way to the lower set of rail stops. Under this condition the entire trackful of wheels on the runway can move down one step nearer the wheel press.

A pneumatic control valve is conveniently located at the end of the press. When a pair of wheels has been demounted and the wheels and axle disposed of, the valve is moved to the proper position and the lower rail stops move away and permit the pair of wheels nearest the press to roll off the runway end into the press. The control valve is then moved to the other position and the lower rail stops move across the rails while the upper stops clear the rails, thus allowing the line of wheel sets on the incline to move ahead toward the press. The rail stops are so spaced that when the lower stops clear the rails, the upper stops prevent another pair of wheels from moving down the incline.

The wheel stop and feeding device which controls the movement of the line of 17 pairs of wheels on the inclined runway consists of an air cylinder and four cam-like stops which move over the rail. In the illustration above the lower stops are across the rails and the upper stops clear

The wheels roll down the incline and are held by the upper rail stops as seen in the upper view at the left. When the two-way valve controlling the mechanism is changed to the other position the lower stops block the rail and clear the upper position, as seen in the lower view at the left. After the wheels have been demounted from the axle the latter slides out and down the chute seen in the view below

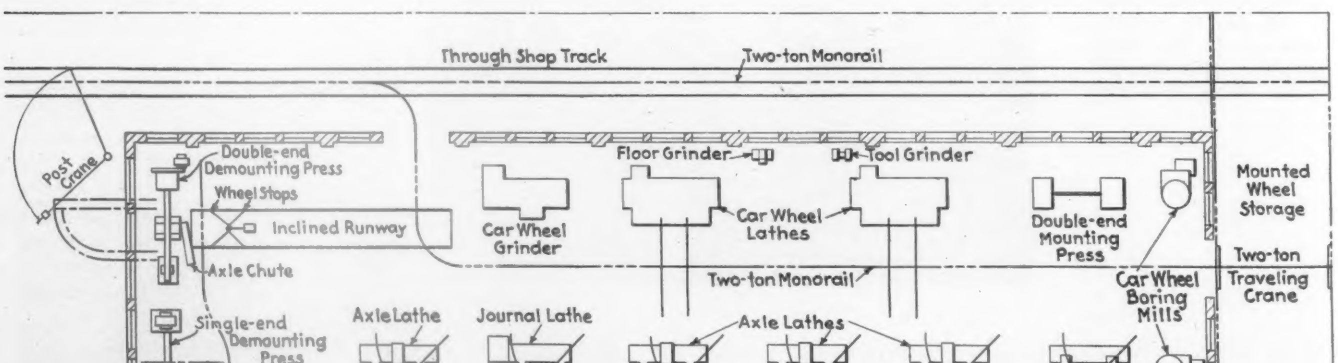


Arrangement of the wheel chutes outside the shop, showing how the two wheels come through the wall openings

When the wheels reach the central point they are picked up by an air hoist and special lifting device

Loading scrap wheels into a car. The hoist controls are handled by the operator in the foreground

When the wheels have been demounted, the two wheels, if they are to be scrapped, drop off the ends of the axle into the chute and roll out through openings in the wall of the shop to a position where they are picked up by an air hoist on a post crane and swung into a scrap car. Demounted wheels which are not to be scrapped are picked up from the press by the overhead hoist and taken to the desired location in the shop. The axle rolls out of the press into a greased chute down which it slides out onto the shop floor. From there it is picked up and



Layout of the Readville wheel shop, showing new demounting facilities

Wheel car used for transporting wheel sets from outlying points—17 pairs make up a car load

taken to the axle lathes, or otherwise disposed of. The illustration shows the arrangement of the scrap-wheel troughs and the guides which direct the single wheels outside the shop. The photographs, showing the outside arrangement, were taken shortly after the new arrangement was installed. During the winter months the outside facilities are covered over as a protection against the weather.

Wheel Handling Cars

Prior to the re-arrangement of the wheel shop facilities it was the practice on the New Haven to load wheels on cars and block them in the conventional manner. One of the illustrations shows a wheel car which has since been designed for the handling of 17 pairs of wheels without the necessity of any blocking. By the use of this car considerable labor and material expense in wheel handling has been eliminated.

Reference to Table II in the March, 1936, issue shows what the Readville wheel shop was doing at that time from the standpoint of output. The table in this article shows the shop performance during six months of 1939. An analysis of the demounting operations will show that it is now possible to demount more than 45 per cent more wheels than was possible with the old facilities.

Air Brake Questions and Answers

D-22-A Passenger Control Valve (Continued)

546—*Q.—What takes place when the packing cup uncovers the slack-adjuster port in the brake cylinder?* A.—When this port is so uncovered, brake-cylinder air flows through the pipe into the slack-adjuster cylinder forcing the slack-adjuster piston outward, compressing its spring. Attached to the piston stem is a pawl extending into the casing, which engages a ratchet wheel, mounted within casing on a screw.

547—*Q.—What happens when the brake is released?* A.—When the brake is released and the brake-cylinder piston returns to its normal position, the air pressure in the slack-adjuster cylinder escapes to the atmosphere through the pipe and the non-pressure head of the brake cylinder, thus permitting the spring to force the slack-adjuster piston to its normal position. In so doing, the pawl turns the ratchet wheel on the screw thereby drawing the cylinder lever slightly in the direction of the slack-adjuster cylinder, thus shortening the brake-cylinder piston travel and forcing the brake shoes nearer the wheels.

548—*Q.—What provides against movement of the ratchet wheel due to vibration?* A.—Ratchet pawls mounted on a floating ring, the end of one of the two levers being held in contact with a tooth in the slack adjuster by a spring.

549—*Q.—What is provided for hand adjustment?* A.—A trip is provided to disengage the holding lever when making a hand adjustment in either direction, a pull of sufficient force being required that, while unlocking is accomplished without undue effort, there is enough movement required and adequate resistance to insure against false movement due to vibration.

550—*Q.—What should be the adjustment if the brake cylinder is mounted on the body?* A.—The piston travel should be adjusted to 7½ in.

Distribution and Sources of Journal-Box Temperature

(*Continued from page 69*)

under conditions of moving air, the velocity of the air being proportional to speed, and blowing against one side of the box in the conventional manner obtaining with the journal box under actual service.

For reference purposes, tests were run with the journal submerged in an oil bath to a depth of one inch, indicated in Fig. 1 as "oil bath only," such tests being designated by this term. The same cycle was used as with the conventional waste pack. A sample comparison, in terms of temperature rise above atmospheric temperature, is shown in Fig. 2 for the temperature at each of the 53 thermocouple locations, as shown by Fig. 1. The temperatures shown in Fig. 2 represent in each case the maximum temperatures obtained after seven hours of continuous running at each respective speed under high atmospheric temperatures.

In general, the center of the journal has the highest temperature; the bearing has the next highest temperature; then, the oil film between the two; and then, the wedge on top of the bearing. The packing temperature is lower than the wedge and lower than the oil-film temperatures. The temperature of the top of the box is lower than the wedge and the waste-pack temperatures.

The comparative temperatures, shown in Fig. 2, are not a measure of the relative proportion of friction losses between perfect bath lubrication and waste-pack lubrication. They have no significance in this respect at all. At 20 m.p.h., the difference in friction loss between bath lubrication and the waste pack was 3.25 per cent; and at 40 m.p.h. it was 1.06 per cent, bath lubrication having the lowest loss. At 60 m.p.h., the difference was 1.6 per cent, the waste pack having the lowest

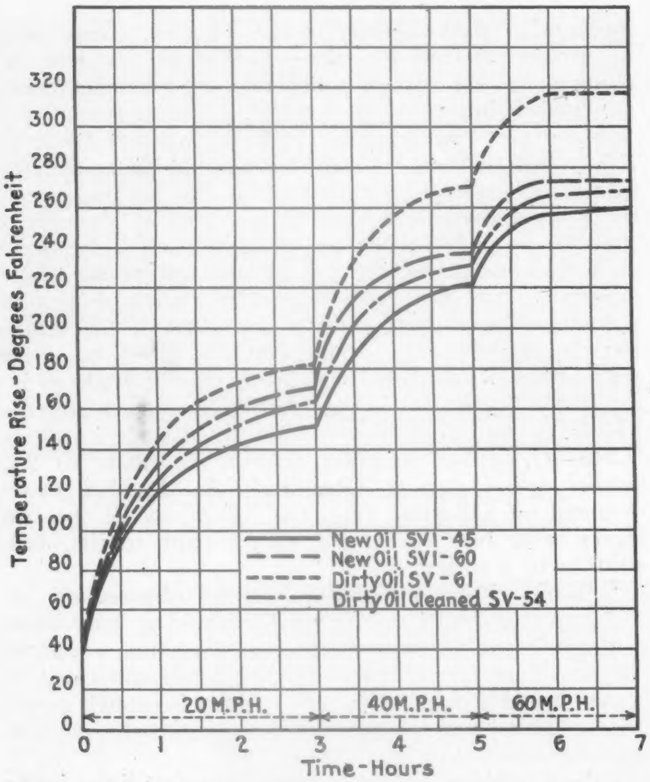


Fig. 3—Temperature at center of journal under no load and without bearing

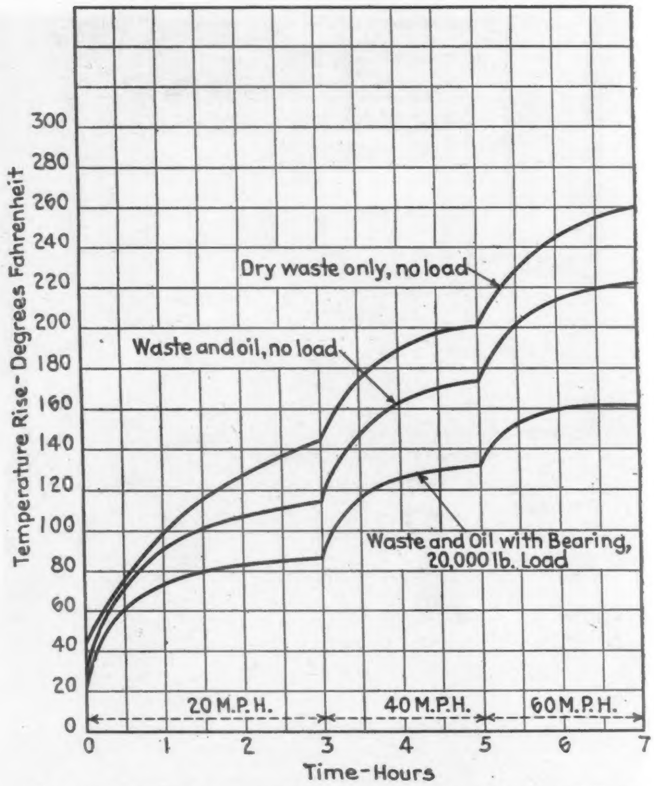


Fig. 4—Temperature at center of journal under various conditions of load and no load

loss. No difference in friction could be expected between the two methods of lubrication, due to the fact that the bearing surfaces were receiving all the oil necessary for successful operation and all other conditions were the same.

Operation in still air and perfect flood lubrication have no practical application in the operation of railroad equipment. It is shown here only to reflect the insulating effect of the waste pack and the effect of air flow against the journal box. Railroad journals to operate at all are immediately subject to air movement. Flood, or bath, lubrication of railroad journals has many mechanical and operating limitations. Due to the fact that railroad journals operate under wide fluctuations of temperature, the insulating effect of waste might be considered an aid to lubrication in its influence on oil viscosity, which would otherwise be controlled by rapid temperature changes, as would exist with flood lubrication. The data reflecting the effect of flood lubrication in still air are to a degree representative of the operating conditions encountered in power-consuming or generating units of a stationary nature.

Effect of Viscosity and Cleanliness of Lubricating Oil on Journal-Box Temperatures

One common factor which influences the amount of heat generated in a journal-box assembly is the viscosity and cleanliness of the lubricating oil. To illustrate this, Fig. 3 reflects the temperature of the center of the journal under no load and without a bearing, comparing the temperature effect of new oils of 45 sec. and 60 sec. viscosity at 210 deg. F., dirty oil as removed from service with a viscosity of 61 sec., and this same oil after cleaning with a viscosity of 54 sec. at 210 deg. F.

The effect of the bearing under load, as a source of increased journal temperature, is not a factor. It will be seen from Fig. 4 that the bearing serves to reduce the operating temperature of the journal by the fact that it aids materially in the dissipation of heat. This is because it provides the principal path for the transfer of the generated heat to the walls of the journal box. This obtains, however, under the condition of clean journal-box packing and with a bearing in proper mechanical condition. Dirty packing or a bearing in improper mechanical condition may be a limiting factor in operation by adding to the heat normally generated in excess above that possible of dissipation through the bearing. The modification of the conventional bearing to compensate further for the effects of temperature and to augment its heat-dissipating capacity has produced outstanding improvements in the actual service operation of bearings.

In Fig. 4 is also shown the temperature of the center of the journal as affected by clean dry waste, and with clean oil-saturated packing, without a bearing or load. Dry packing, or packing with a reduced oil content, approaches the temperature effect of the dry waste shown in proportion to the oil deficiency, which would be further augmented by increasing the dirt content of the packing, as reflected by the combined evidence of Figs. 3 and 4. In actual operation this condition is a source of heating, which is due to the effects of service, producing undersaturated dirty packing.

Conclusion

Obviously, the control of heat sources is the first step in avoiding the limiting effects of excessive temperature rise in the conventional journal-box assembly. The conditions of the waste, the oil, and the bearing are the basic factors in controlling temperature sources. There is no economic substitute for clean journal-box packing properly saturated. The proper conditioning of bearings at the time of initial installation, and their modification in design to compensate for the effects of temperature, increase the factor of dependable operation. The extensive reduction to practice of these fundamental principles by several railroads has amply demonstrated their value.

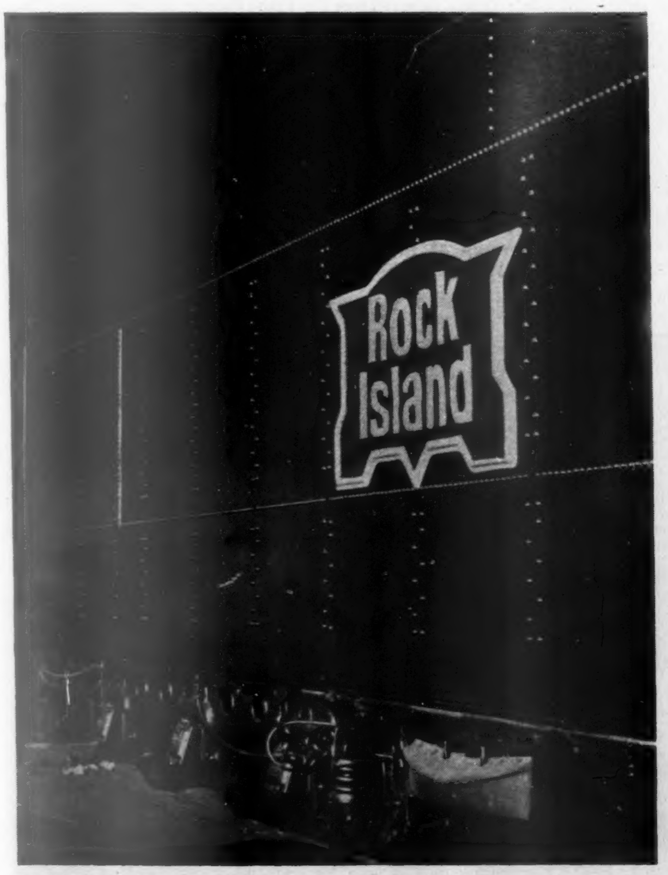
IN THE BACK SHOP AND ENGINEHOUSE

Rock Island Medallion

The medallion, illustrated on the side of a locomotive tender of the Chicago, Rock Island & Pacific, is not only a highly effective and attractive nameplate, but has a number of other rather important advantages. In the first place, from the point of view of durability, this medallion will easily have a service life two or three times that of the ordinary stencilled letters usually applied to locomotive tenders. Then again, the medallion is easily taken from the tender side by simply removing a few cap screws and can be as easily reapplied after any necessary repainting or repair work has been done.

The medallion is made of 16-gage, hot-rolled, annealed, pickled steel. It is 44 in. by 63 in. in size and has especially embossed edges and raised letters. The extreme edges are more deeply embossed than the main background of the medallion which is spaced about ⅝ in. away from the tender side plate so as to leave room for the thin heads of a suitable number of ½-in. cap screws which are welded to the tender plate. The screw ends project through holes in the medallion which is held firmly in place by ½-in. hex-head brass cap nuts.

The back of the medallion is finished with a durable rust-resisting aluminum enamel, while the face is finished with high-baked Dulux build-up, consisting of a primer, aluminum and red colors, and clear varnish. The cap-screw holes are punched at proper points around the edge just inside the embossed moulding, not being clearly shown in the illustration. The background is red and the edge and letters are finished in aluminum.



Rock Island locomotive tender with embossed medallion which is attractive and durable and may be easily removed or reapplied if necessary

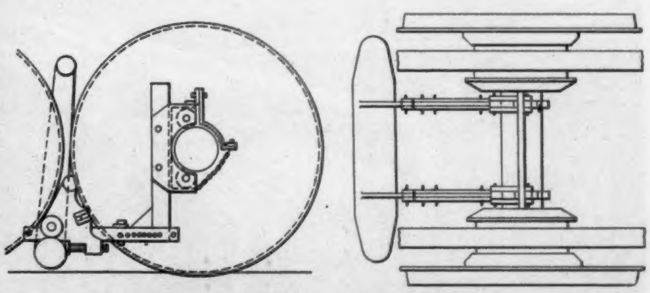
Tire-Turning Tool Has Cutter Control

A method for machining tire flanges and trimming the edges of tires on locomotives without removing the wheels, by the use of a dummy brake head acting as a tool holder, has been used for years, but the lack of control of the cutting tool with this method has limited its use to these operations. The apparatus shown in the diagram has been developed and patented by F. L. Hall, Cincinnati, Ohio and has an arrangement for controlling the cutting tool by utilizing the axle as a center thus making it possible to turn tires to a true circle.

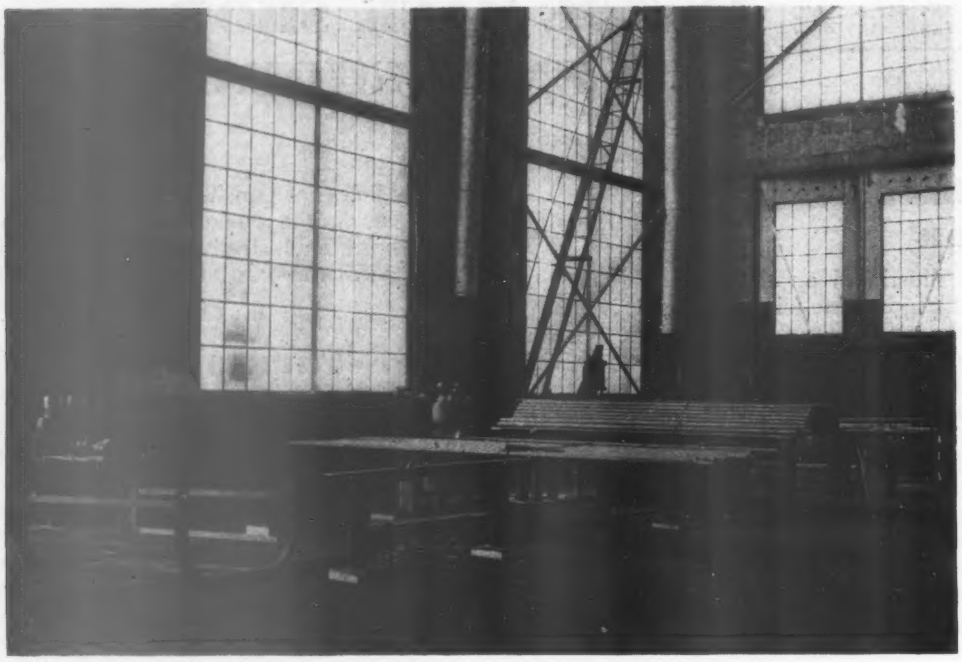
A tool holder with a cutting tool is substituted for the brake head and the brake shoe. Two centering devices are applied to each axle inside the driving wheels with their extensions running back either over or under the brake beam. A brake-beam stop is set in the center of the extension and is adjusted to allow the tool holder to clear the tire when braking pressure is applied. The cutting tool can be located in the tool holder for the depth of cut by setting it with the outside edge of the tire. A round-nose tool can be used for a roughing cut and rough flanging tools are suitable for reducing high flanges, after which a tread-contour tool may be used for the finishing operation.

When braking pressure is applied, the tool becomes engaged with the tire. Then the locomotive is towed by another locomotive or by some other outside source of power. The braking pressure is regulated by adjusting the safety valve on the distributing valve. The brakes are controlled from the ground by means of a rope extending through the cab window.

One hour per pair of driving wheels is required to apply the device while 30 minutes per pair of driving wheels is needed to remove the device and apply the brake heads, although this time may be reduced by an experienced operator having available the proper pit facilities. The tires of over 100 locomotives have been turned with this apparatus at three railroad repair shops. As an example, it is claimed that the cutting time was 12 hours for removing from 3/32 in. to ¼ in. of metal from the tires of five pairs of driving wheels on one locomotive.



A dummy brake head acts as a tool holder and the axle is used as a center in this arrangement for turning tires without dropping the wheels



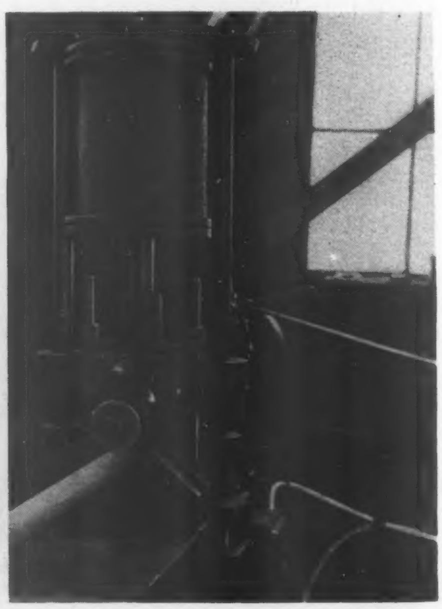
Flue Repair Shop Designed For One-Man Operation

On a small road the problem of efficient and adequate repair facilities is sometimes more difficult to solve than on a large road for the reason that a small volume of work does not readily fit into the commonly accepted ideas of production methods. At Pen Argyl, Pa., the Lehigh & New England has developed its locomotive repair-shop facilities on a basis of providing shop equipment designed for low production costs on a low volume of work. The flue shop described in this article is a typical example of one of the shop departments the equipment of which can be used with a minimum expenditure for labor.

The big-shop supervisor, who thinks in terms of maximum output per day, can appreciate the problem of this shop when it is stated that the requirements for locomotive flues and tubes at the Pen Argyl shop averages about 150 a month under present conditions. (In describing these operations the term flue is hereafter used to include all sizes, from 2 in. to 5 in.) The flue shop, however, when operated with more than one man, has a capacity considerably greater than this if a demand occurs.

Reference to the flue department layout, shown in Fig. 6, will indicate that the work space of the department is about 28 ft. by 55 ft. and that it is designed for one- or two-man operation. The flues, after being re-

Fig. 1 (top of page) shows the general arrangement of the flue shop looking from the end of the sloping runway where the finished flues drop into the cradle—The swedging hammer and heating furnace are seen at the left—An operator is working at the welding position and the top of the cutting machine is silhouetted against the window, above the pile of flues—Fig. 2 (below, left) shows the flue-cutting machine with its adjustable length stop and centering device, the roller support and the cutting head at the far end—Fig. 3 (below, right) is a close-up of the cutting head showing the air cylinder, roller drive and controls for air and electricity.



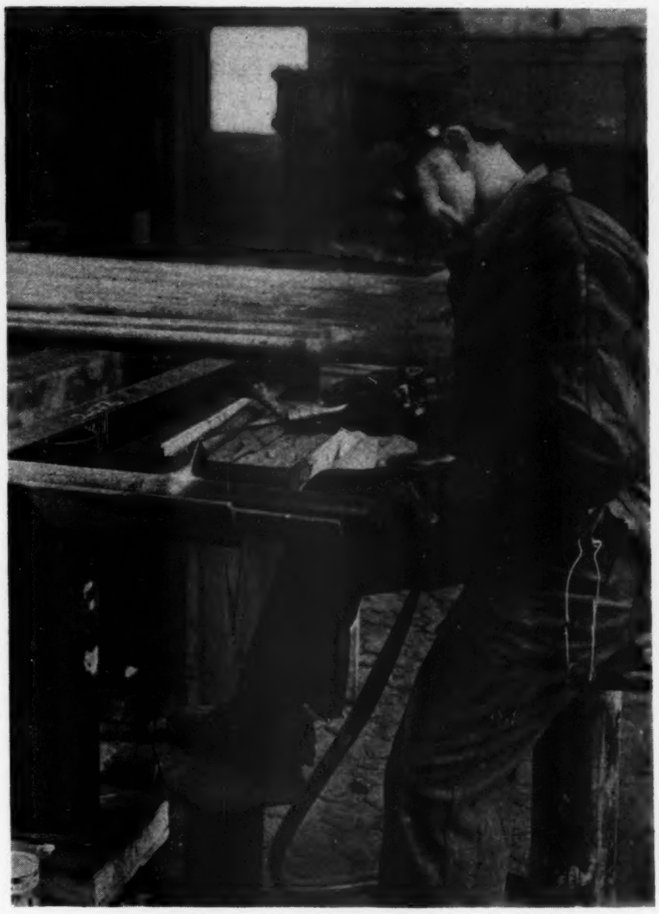


Fig. 4 (above)—The swedging hammer in operation.
Fig. 5 (right)—The flue and the safe end are laid in the angle-iron trough, then welded by the oxy-acetylene process and finally annealed.

moved from the locomotive and cleaned in a rattler outside the shop, are placed in a cradle and brought into the shop through a door adjacent to the flue cutter. They are picked up by an overhead traveling crane and placed on the high end of a 55-ft. double-rail runway which, by means of properly located sloping portions, causes the flues to roll by gravity from one operation position to another until finally they drop into a cradle at the low end of the runway. The total drop of the runway, in its 55-ft. length, is 12 in.

Cleaned, unrepaired flues are picked up from the runway rails and cut off to the proper length. Fig. 1

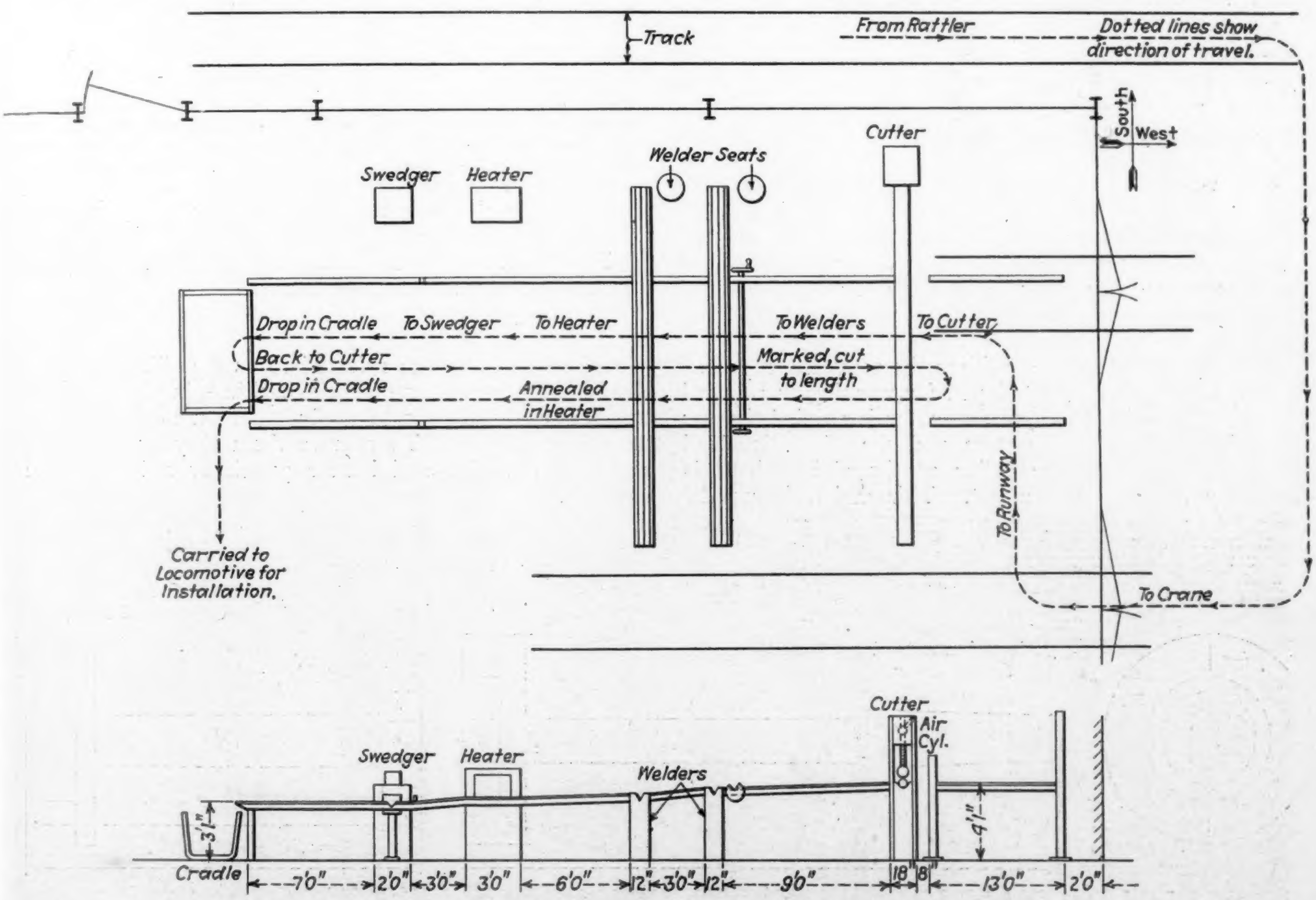


Fig. 6—General arrangement of the flue shop and an elevation of the sloping runway

shows the general arrangement of the repair facilities and Figs. 2 and 3 show the details of the flue-cutting machine. It will be noticed, in Fig. 2, that there is a slot in the inverted-channel table of the cutting machine in which a roller flue-end centering device is secured. This device not only serves to hold the end of the flue in the proper position to assure a cut at right angles to the center line of the flue but may also be adjusted to control the accuracy of the length of the flue.

Fig. 3 is a closer view of the cutter head. The cutting rollers are driven by belt from a motor and pressure on the cutters is applied by means of a pneumatic cylinder. Air and electric controls are located so that they are convenient to the operator. The roller support, which can be seen near the centering device in Fig. 2, simplifies the act of slipping the end of the flue over the mandrel of the centering device.

When the flue is cut it is placed on the sloping runway rails and it rolls down to the safe-end welding position. The drawing shows a flue-stop device at the welding position which had not been installed when the photographs were taken.

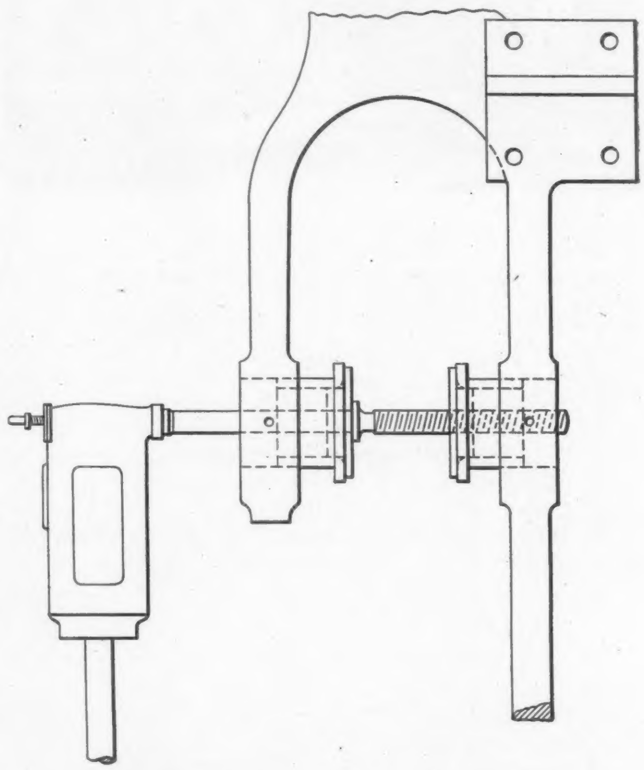
The welding position consists of angle-iron troughs into which the flue drops. Here the safe end is dropped into the trough with the flue and is welded to the flue end by the oxy-acetylene process. After the welding is completed the flue is rolled down the runway to the heating furnace position where the bead of the weld is hammered down and the end swedged. The flues are then dropped into the cradle and returned to the cutting position where they are marked and cut to length. They then roll down the runway to the heating furnace where they are annealed. After the annealing is done the flues again drop into the cradle at the end of the runway and are ready to be transported to the erecting shop for application to the locomotive boiler.

Applying Link Trunnion Bushings

By A. D. Hollis

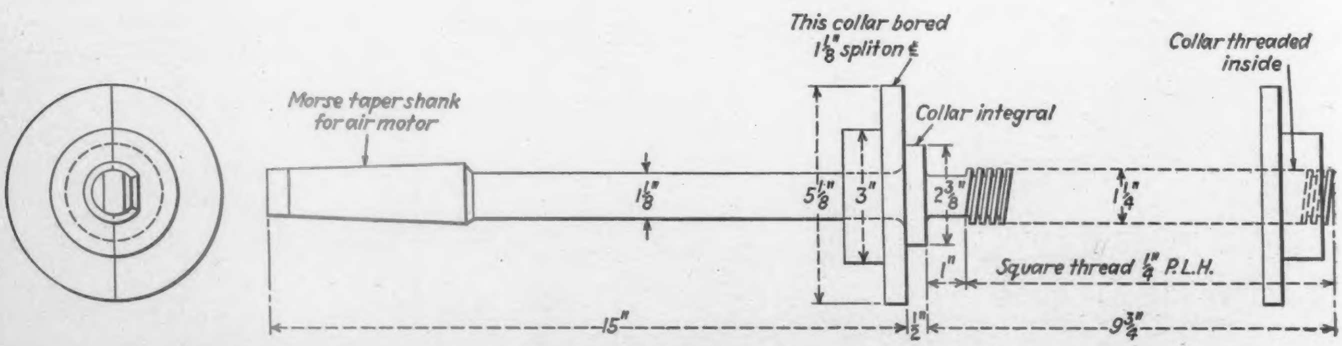
A convenient device which utilizes screw action and the power of an air motor to apply link-trunnion bushings in Walschaert valve-gear brackets is shown in one of the drawings and the use of the tool in the second drawing. The device is unique in that it forces in both bushings at the same time and utilizes the convenient type of close-quarter generally employed for reaming in railroad shops. When the power of the motor is applied the bar turns in a clockwise direction and the ¼-in. left-lead screw backs out of the collar at the right which is internally threaded like a nut. This action forces the bushings in place. The collars are shouldered to fit freely inside of the bushings used on all locomotives equipped with Walschaert valve gear. The collar at the left is bored a free fit on the 1⅛-in. portion of the bar and is split on the center line so that it may be removed in two halves.

The use of this device tends to spread the two members of the link bracket apart slightly, so, after placing a good strain on the bracket, the motor is shut off and the bracket struck on the outside with a copper sledge hammer. In this manner the bushings may be pressed in without spreading the bracket. On some light types of link brackets that are not well supported it may be desirable to use a bolt and clamps so the bracket will not be spread enough to take a permanent set and thus be thrown out of alignment.



Link trunnion bushing applicator set up ready for use

Although sometimes referred to as a pulling bar, the threaded bar actually pushes the bushings in place. With this tool it is possible to apply trunnion bushings in a very few minutes. To remove them they are knocked out with a sledge and handle punch or a long-stroke hammer.



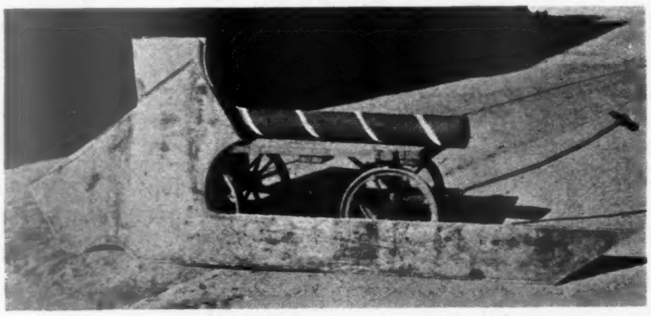
Detailed dimensions of a device used in applying Walschaert link trunnion bushings by screw pressure

Syphon Patch With a Spiral Seam*

The completed syphon patch, shown on the truck in the illustration, was fabricated from a sheet of a shape similar to the sheet which is leaning against the truck. Four syphon patches of this type were applied to syphons of two locomotives on a western railroad.

This patch was designed and constructed with a spiral seam because of the lack of equipment for making a patch of this size with a straight seam running the length of the patch, a distance of 84 in. A sheet was cut to the shape of the developed surface of that part of the syphon which the patch replaced. It was formed by rolling and then welded along the spiral seam.

* An entry in the prize competition on boiler patches, announced in the March, 1939, issue. The names of the winners were published in the August, 1939, issue.

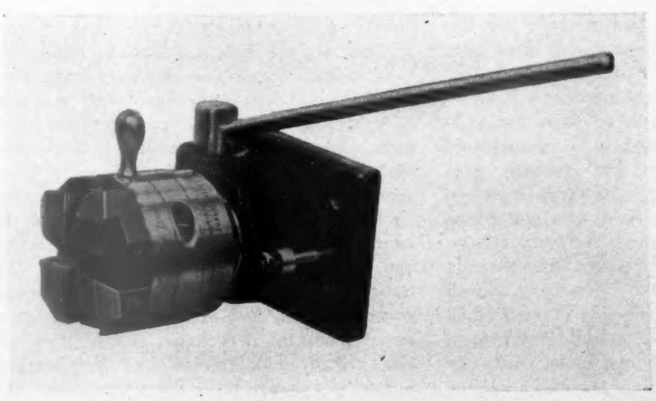


The completed syphon patch is shown on the truck—The shape of the patch before rolling is indicated by the sheet leaning against the truck

Hand-Feed Adapter For Turret-Lathe Head

When using a die head on a heavy-duty turret lathe, a mechanism of some type is recommended which will enable the operator to relieve the head of the load of the heavy turret and carriage. A sliding adapter designed by the Landis Machine Co., Waynesboro, Pa., for use with Landmatic heads not only permits of a floating action for the die head, but also provides a gear mechanism for starting the die head onto the work by hand.

The usual type of sliding adapter provides a floating action which permits the operator to advance the turret independent of the head, but no provision is made for starting the die head onto the work other than by means of the rack and gear which are ordinarily used to advance the carriage. The forward or thread-starting movement of the die head in the Landis adapter is accomplished through an intermediate gear mechanism within the adapter. A rack gear is milled axially along the peripheral surface of a bushing to which the die head shank is fitted. A pinion gear, whose teeth engage the rack gear on the bushing, is fitted with a long handle which is used to actuate the movement of the bushing.

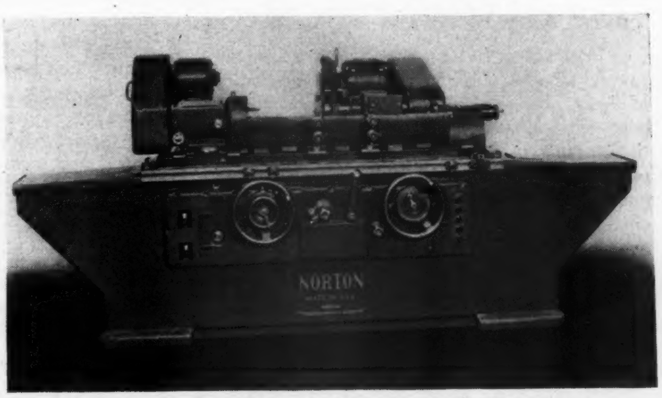


Landis hand-feed adapter for Landmatic heads

Universal Machine for External and Internal Grinding

A universal grinder, which performs both external and internal grinding, has been designed by the Norton Company, Worcester, Mass. This machine, known as the type LC Multipurpose, has a swing of 12 in. and is built in 24, 36, 48, and 72-in. lengths. It requires a floor space about 5 ft. in width and a length of from 9½ ft. for the shortest machine to 17½ ft. for the longest machine.

The base is a ribbed single casting with the reservoirs for the coolant and the oil for hydraulic operation cast as an integral part. The table can be propelled hydraulically or by hand through a two-speed arrangement, thus making it possible to move the table quickly into position or more slowly for shoulder grinding or similar operations. A single lever disengages the rapid hand traverse and engages the slow traverse. An electric dwell control for the hydraulic traverse mechanism is furnished.



The Norton 12-in. type LC Multipurpose grinder with the splash guards in place

The grinder has two ranges of feed, either of which may be selected by pushing in or pulling out a single knob. Each hole in the index is equivalent to a work diameter reduction of .004 in. in the fast range and .0001 in. in the slow range. The headstock is of the universal type and is driven by either a ½-hp. a.c. constant-speed motor or by a variable-speed d.c. motor. When an a.c. motor is furnished, four work speeds ranging from 65 to 260 r.p.m. are obtained by cone-type V-pulleys. A push button controls the work speed.

Both live-spindle and dead-center operations can be performed, the change from one type of drive to the other requires only the adjustment of a knob on the front of the headstock. The base is graduated and can be set at any desired angle either side of the zero position. All bearings are pressure lubricated. The final drive to the face plate is by a chain.

High Spots in

Railway Affairs . . .

How People Travel In This Country

The Interstate Commerce Commission, in its annual report to Congress, makes some interesting observations as to the division of passenger traffic in this country in 1938 between the various forms of transport. Leaving out of consideration urban traffic, it is estimated that the private automobiles carried 84.95 per cent of the intercity travel. This is a long step from the horse and buggy days. The steam and electric railways carried 8.44 per cent; the busses, 5.87 per cent; rail and waterways, .56; and the air carriers, .18 per cent. The passenger-miles by air amounted to about 6 per cent of the pullman travel. If the private automobile is eliminated from the calculations, and only the common carriers are considered, then the railroads accounted for 56.1 per cent of the passenger-miles in 1938, the busses, 3.90 per cent; the rail and waterways, 3.7 per cent; and the air lines, 1.2 per cent.

Taxes, And More Taxes

The railway stockholders have taken a severe licking all during the '30's. Some of them, preferred as well as common, have been entirely wiped out in the reorganizations—and even the bondholders have not gotten off any too well. The governments, however,—federal, state, county and municipal—have had a generous slice of the railway's earnings. The Railway Age points out that, "for every dollar of net income earned by the railways for their stockholders in the nine years ending with 1939 their taxes amounted to $12.38. During this period taxes averaged more than $295,000,000 annually, while net income averaged less than $24,000,000. The total taxes paid in the years 1931-1939, inclusive, amounted to $2,656,056,000; net income earned in these same years was $214,564,000. Thus the pay-off of the tax collector, as compared with the owner, was more than twelve to one, with the tax collector on the long end of the transaction."

"Orphan Annie"

Nobody in official Washington seems to be greatly concerned about relief to the railroads in the form of more equitable regulation. Bills, designated as S. 2009, somewhat similar in contents, but quite different in form, were passed by both the Senate and the House before adjournment last summer. They were sent to conference and the expectation was that the conferences would get together in the fall or early winter and have a report ready to present early in the present session. They did not do so, and it is not apparent at this writing just when they expect to "get down to brass tacks." The conference committee is scheduled to hold a meeting February 1. The President's message was concerned mostly with foreign affairs and national defense, although in enumerating "other items of great public interest" he did list "the freeing of large areas from restricted transportation discrimination," whatever that may mean.

President Roosevelt On Waterway Tolls

"Uncle Sam," contrary to New England tradition, has been quite a spendthrift in these recent years. Apparently he is beginning to get worried about the day of reckoning. Said President Roosevelt in his recent budget message to Congress: "I have always believed that many facilities made available to our citizens by the government should be paid for, at least in part, by those who use them. * * * A start on this policy has been made. In such a way a substantial part of the annual cost of maintenance of roads, trails, and grounds in forests and parks will come back to the Treasury and reduce the annual cost of government. Another example is the $50,000,000 the government spends annually in the maintenance of dredged channels, buoys, lighthouses, lifesaving stations, and so forth. It would seem reasonable that some portion of these annual expenditures should come back in the form of small fees from the users of our lakes, channels, harbors, and coasts."

The Struggle For Freight Business

The jurisdiction and operations of the Interstate Commerce Commission have been considerably broadened in recent years because of the rapid development of intercity common-carriers other than the railroads. This is reflected in its annual reports. In the current one, recently presented to Congress, it attempts to give an overall statistical picture of the operations of these carriers, although it freely admits that its estimates may have a considerable margin of error. Of the intercity traffic on a ton-mile basis the railroads carried 62.82 per cent in 1938; the inland waterways, 14.33 per cent; the oil pipe lines, 14.26 per cent (it was only 11.97 per cent in 1937); the private intercity trucks, 4.83 per cent; and the intercity tracks for lines, 3.76 per cent. The ton-miles on inland waterways were much lower in 1938 than in 1937, because of a heavy falling off in ore tonnage on the Great Lakes. The railways are certainly faced with plenty of competition.

Good Year for Scrap

War—whether declared or not—always seems to have a favorable effect upon the scrap market. Much scrap was shipped from this country to Japan in the last two years, and there has been a large demand for it in the United States since the steel industry speeded up its production in recent months. It is estimated that last year about 70 per cent of the scrap went into the production of new steel, 20 per cent into castings and 10 per cent into miscellaneous uses. The Railway Age estimates that in 1939 the railroads sold about $55,000,000 of iron and steel scrap, in addition to which they received approximately $25,000,000 in freight revenue for transporting scrap. This revenue compared to about $14,000,000 in 1938; neither year, however, approached anywhere near the traffic revenue from scrap in 1937 of $34,060,000.

Returning to Sanity

It is rather interesting in these days to wander about behind the scenes in Washington and study the reactions of those in high places. Whatever the cases may be—the approaching election, the war clouds hanging low over the world, a colossal and still mounting public debt, New Deal policies on the defensive, or what not—there is quite a different atmosphere than that which existed a couple of years ago. Indeed, there has been a steady shifting for many months, away from cocksure and dogmatic statements, all along the line. Exponents of radical measures, who formerly spoke in no uncertain tones of conviction, are much less outspoken today, and even admit that some mistakes have been made—an almost unheard-of and unthinkable attitude on their part a few years ago. The "wise boys on the Hill"—members of Congress—who have been in touch with their constituents back home reflect this swing away from the left toward the center in even more decided fashion. One can almost imagine them saying: "We have had our fling. Let's get down to sound common sense and see what we can conserve from the wreckage, before it is too late. Many good things have been started, but let's separate them from the follies and discard the latter."

(Turn to next left-hand page)

Have you Seen . . .

THE STORY OF THE CHILLED CAR WHEEL

This talking picture shows how Chilled Car Wheels are made, portrays the work of our Association Inspection Department and illustrates the operations carried out in our Research Laboratory — one of the finest in the country.

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Among the Clubs and Associations

Car Foremen's Association of Chicago.—Meeting February 12, 8:00 p. m., LaSalle Hotel, Chicago. Speaker: J. E. Mehan, assistant to superintendent of car department, Chicago, Milwaukee, St. Paul & Pacific. Subject: Discussion of changes in the A. A. R. Rules of Interchange.

Southern and Southwestern Railway Club.—Meeting March 21, 10 a. m., Ansley Hotel, Atlanta Ga. Motion picture of the Tennessee Coal, Iron & Railroad Company on the "Making and Shaping of Steel."

Northwest Car Men's Association.—Meeting February 5, St. Paul, Minn. Speaker: L. R. Kassick, general foreman, Soo Line, Minneapolis, Minn. Subject: Proposed changes in A. A. R. rules as presented by A. A. R. committee.

Toronto Railway Club.—Meeting February 26, 7:45 p. m., Royal York Hotel, Toronto, Ont. Speaker: T. V. Buckwalter, vice-president, Timken Roller Bearing Company, Canton, Ohio. Subject: Steam Locomotive Slipping Tests and Photoelastic Study of Stresses on Railway Axles. Motion pictures.

Central Railway Club of Buffalo.—Meeting, February 8, Hotel Statler, Buffalo, N. Y. Speaker: L. W. Horning, regional director, Association of American Railroads, New York. Subject: New Frontiers in the Railroad Industry. Central Railway Club chorus.

Railway Club of Pittsburgh.—Meeting January 25, Fort Pitt Hotel, Pittsburgh, Pa. Speaker: Dr. Harvey Bartle, chief medical examiner, Pennsylvania, Philadelphia, Pa. Subject: Medical Side of the Railroad Industry and Its Management.

Canadian Railway Club. — Meeting, February 12, 8:15 p. m., Rose Room, Windsor Hotel, Montreal, Que. Speaker: A. Reyburn, foundry superintendent, Canadian National. Subject Mechanization of Foundry. Illustrated.

New England Railroad Club.—Meeting, 6:30 p. m., February 13, Hotel Touraine, Boston, Mass. Speaker: E. D. Campbell, general mechanical engineer, American Car and Foundry Company. Subject: Riveted and Welded Construction of Freight and Passenger Cars; illustrated by lantern slides. Dinner prior to address.

Club Papers

Diesel Switchers Show Savings

Pacific Railway Club.—At the September meeting of the Pacific Railway Club at Los Angeles, Calif., some specific information regarding the performance of Diesel locomotives in switching service was presented by M. D. Raymond, Pacific Coast manager of the American Locomotive Company. ¶Any new type of motive power, said Mr. Raymond, must justify itself by showing a net return on the investment. That the Diesel switching locomotive can justify itself on this basis can easily be substantiated by a careful study of some 350 odd installations now operating on American railroads. These savings can be divided into two classes: first, the direct savings in operation against the present-day operation of steam switchers, and second, the indirect savings. ¶The direct savings are based on the differences in cost between steam and Diesel operation, considering the following items: wages, fuel, water, lubrication, maintenance, enginehouse, and miscellaneous. The first item covering wages of the engineman and fireman will be the same in both cases, whether Diesel or steam. The principal savings come from fuel, water, maintenance, and enginehouse expense, the savings in enginehouse expense being mainly in boiler washing, hostling, fueling, and fire-cleaning. Based on figures obtained from a great many railroads, this saving runs all the way from $1.75 to $2.50 per operating hour. Taking the minimum figure of $1.75 an hour, and considering an average continuous switching operation requiring 7,200 hours per year of service, it means that there is a direct saving of $12,600 per year by the use of Diesel switching power. ¶The indirect saving depends greatly upon the individual case and whether or not it is possible to abandon coal-dock and ashpit facilities, and boiler-washing equipment. Full benefit, therefore, cannot be derived from long engine runs unless the steam switchers are entirely replaced at these intermediate terminals with some other form of power which does not require the use of the facilities mentioned. Since the Diesel does not require any of the special facilities demanded by the steam switcher, it is ideally suited for this purpose and its installation at these terminals will add materially to its direct savings. ¶A true picture of the economic side involves consideration of the cost of Diesel locomotives. A price reduction averaging about 10 per cent has been effected in the past year. So-called stock designs can be produced in quantities resulting in substantial reductions in manufacturing costs which in turn can be passed along to the railroads. This work has progressed to such an extent now that the added cost of a modern Diesel switcher is only approximately 1.4 times the cost of an equivalent steam switching locomotive. Since the Diesel switcher will replace 1.4 steam switchers, the total investment for new power would be no greater for Diesel switchers than it would be if new steam locomotives were purchased. ¶From the point of view of direct savings alone, it can be readily seen that the Diesel switching locomotive will pay for itself in a maximum period of from five to seven years. In fact, there are many instances where these Diesel units have paid for themselves in three years' time. ¶In discussing Mr. Raymond's paper, Lee Pearson, road foreman of engines of the Atchison, Topeka & Santa Fe at San Bernardino, Calif., said that too many cars are frequently handled in switching movements and the resultant run of slack in high-speed switching, with either Diesel or steam power, causes damage to cars and lading which may not be apparent at the time. Road handling is also to blame for some damage, but the preponderance of the damage is done in yard handling. ¶In operating switch locomotives, therefore, whether Diesel or steam, necessary care must be taken to control slack. Enginemen are instructed, in operating their locomotives, regardless of the kind of power being used, to apply the brakes lightly when a stop signal is given and let the slack run as gently as it will. After they feel the "tug" of the tonnage in the cut they can then apply the brakes to their full value and average good handling will result. ¶Mr. Pearson said that the necessary power to turn the traction motors assists considerably in retarding the speed while drifting up to a cut of cars, this retardation being much more pronounced with a Diesel switcher than would be the case with a steam switching locomotive. However, a few days of experience in operating Diesel switching power usually enables the engineman, with one operation of his throttle, to come up to a cut of cars at the desired speed and make a coupling without any appreciable or, at least, any objectionable shock. ¶The rough handling mentioned is particularly obnoxious when handling a cut of occupied passenger cars, but, in Mr. Pearson's opinion, only a week or ten days of experience is required for the average engineman to lose his fear of handling the Diesel switching locomotive and become very proficient in its operation and handling. He said that wonderful results have been secured in Diesel switching service when the operation and handling are fully understood, and that substantially less rough handling will unquestionably be brought about in yards by the more general adoption of Diesel switching power.

(*Turn to second left-hand page*)

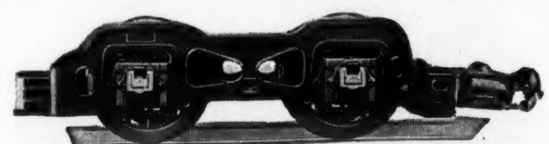
THE rated draw bar pull of a locomotive is the average of the constantly varying effort exerted throughout one revolution of the driving wheels. Such a rating considers that the locomotive is moving and is by no means a correct indication of the power exerted in starting from a state of rest.

Starting power depends entirely upon the exact position of the cranks at the instant of starting. Crank position not only controls the leverage between piston and driving wheel but also determines whether or not both valves are open for admission of steam to the cylinders.

The above chart illustrates a typical starting draw bar pull curve for a modern two-cylinder locomotive with Booster. Curve "A" shows the starting draw bar pull of the main locomotive for all crank positions. Curve "B" represents the combined starting draw bar pull of the locomotive and Booster, the shaded area between curves "A" and "B" being the starting draw bar pull added by the Booster. Curve "C" in the lower portion of the chart shows in percent the increased power with the Booster for all crank positions.

The chart also clearly shows that for about 216° or 60.4% of the crank circle the power of only one cylinder is available for starting. This means that most starts would be made with one cylinder, or slack would have to be taken to place the cranks in a more favorable position, if Booster power is not used.

It will also be observed that at four points on the crank circle the starting effort of the locomotive reaches a minimum at which approximately one-third of the rated draw bar pull is exerted. It is at these low points that Booster power is most effective, in some cases adding as much as 67% to the starting draw bar pull of the locomotive, or the equivalent of 2 additional driving axles.

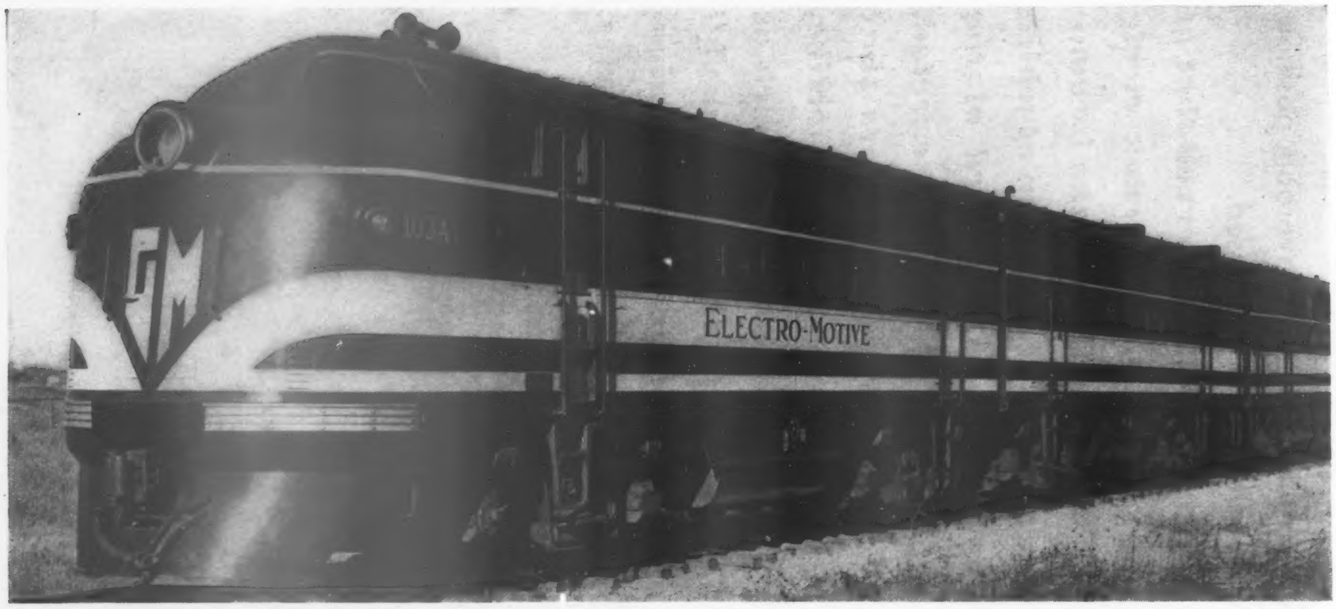


*Trademark Registered United States Patent Office

REG. U.S. PAT. OFF.

FRANKLIN RAILWAY SUPPLY COMPANY, INC.

NEW YORK CHICAGO MONTREAL



Two Electro-Motive Diesel-electric freight locomotives coupled to form a 5,400-hp. unit—Each locomotive consists of two units each containing a 16-cylinder, 1,350-hp. General Motors Diesel engine—Now being tested on the Atchison, Topeka & Santa Fe

NEWS

Lawford H. Fry to Lecture at Franklin Institute

LAWFORD H. FRY, railway engineer, Edgewater Steel Company, Pittsburgh, Pa., will lecture on The Steam Locomotive, Its Development and Present Position in Railroad Transportation, at the Franklin Institute, Philadelphia, Pa., on February 21.

Arch-Bar Truck Rule Modifications

ACCORDING to Circular D. V.—972, recently issued by the Association of American Railroads, Mechanical division, certain modifications have been made in the interchange rule which prohibit acceptance from owners of freight cars equipped with arch-bar trucks after December 31, 1939.

Effective January 1, 1940, a note is added to Par. (4), Sec. (t), Rule 3, as follows: "(t-4) Trucks, with arch bars, prohibited, on and after January 1, 1940, under all cars. From Owners.

"Note.—The movement of cars equipped with arch-bar trucks must be confined to owner's rails, except that they are acceptable in interchange from owner for loading or for unloading within the same terminal switching district in which the interchange occurs, provided that no road haul is involved in such movement, and provided that cars so interchanged will be immediately returned to owner's rails when loading or unloading is accomplished.

"Cars equipped with arch-bar trucks are acceptable for movement between plants located in the same switching district, provided no road haul is involved."

Effective January 1, 1940, the second note following Par. (2), Sec. (w) Rule 3, is modified to read as follows:

"Note.—Industrial or other cars not intended for interchange service, when moving on their own wheels, may be accepted in interchange in their initial movement from manufacturer to destination (or seaboard) without meeting the requirements of Sec. (a), Par. (1), in so far as the retaining valve and A. A. R. standard triple valve are concerned, second paragraph of Sec. (b) for No. 2 A. A. R. brake beams, first, second and seventh paragraphs of Sec. (c), third paragraph of Sec. (s), paragraphs (1), (2-a), (2-b) and (4) of Sec. (t) and Specifications for Tank Cars. To each side of such cars a card shall be attached by shippers, reading as follows: 'Industrial or Export Car shipped in accordance with A. A. R. Rule 3. Signed............Shipper.'"

Industrial Progress Award Program

THE James F. Lincoln Arc Welding Foundation has announced a 2½-year program of scientific study for improving designs, manufacture, fabrication, construction, and maintenance of all types of machines, building, structures and products which will culminate on June 1, 1942, in the payment of $200,000 in awards, ranging from $13,700 for first prize down to $100. Locomotives, freight cars, passenger cars, and locomotive and car parts are in Classification C, Divisions C-1 to C-4, inclusive.

Inquiries for further information concerning the program should be addressed to the Secretary, The James F. Lincoln Arc Welding Foundation, Cleveland, Ohio.

Winterrowd Elected to Board of Managers of Franklin Institute

W. H. WINTERROWD, vice-president in charge of operations of The Baldwin Locomotive Works, has been elected a member of the Board of Managers of The Franklin Institute, Philadelphia, Pa., a representative group of industrial, financial, and educational leaders who direct its affairs. The Institute is devoted to the advancement of science and the promotion of the mechanic arts.

Annual Report of the Bureau of Safety

MAKING his last annual report as director of the Bureau of Safety, Interstate Commerce Commissioner William J. Patterson reviewed the fiscal year ended June 30, 1939, in a 52-page pamphlet setting forth in the usual form results of inspection of safety-appliance equipment on railroads, together with information concerning the hours-of-service records of employees, installations of signals, investigation of accidents, and other activities of the Bureau. Mr. Patterson who had been director of the Bureau of Safety since March, 1934, became a member of the commission by appointment from President Roosevelt last July; and Shirley N. Mills succeeded him as head of the Bureau.

During the year under review a total of 1,144,168 cars and locomotives was inspected, 29,232 or 2.55 per cent were found defective, as compared with 2.41 per cent defective out of the 1,213,081 inspected in 1937-38. The percentage defective in 1938-39 (the above-mentioned 2.55 per cent)

(Continued on next left-hand page)

was greater than in any of the preceding nine years, and compares with a post-1930 low of 1.83 per cent in 1931-32.

Air-brake tests were made on 2,508 trains, consisting of 110,467 cars, prepared for departure from terminals; air brakes were found operative on 110,390, or 99.93 per cent, of these cars. This percentage, however, was attained only after 753 cars having defective or inoperative brakes had been set out, and repairs had been made to the brakes of 700 other cars in the trains. According to the report, it was found necessary to repair the brakes or set out an average of nearly three cars for every five trains. Tests on 805 trains arriving at terminals with 39,979 cars showed that the air brakes were operative on 98.55 per cent of the cars—the cars with inoperative brakes averaging slightly over two for every three trains tested.

Commenting on the program for equipping cars with AB brakes, Mr. Patterson reports that improvement is not being effected as scheduled; that "appropriate action to expedite this improvement is required." During the year under review 43,234 additional cars were equipped, bringing the total to 13.64 per cent of the number of cars used in interchange service. This 13.64 per cent is a composite figure, representing 14.47 per cent of the railroad-owned cars and 8.27 per cent of those owned by private car lines. Meanwhile 4½ years or 45 per cent of the 10-year-period allotted for making the improvement had elapsed on last June 30; and "a considerable number of car owners have not even started . . ., the equipped cars . . . having been reported by only 88 railroads and 77 private car lines." The Bureau next observes that further tests are required for final determination of the proper cleaning period for AB brake equipment, now fixed tentatively at 36 months by the Association of American Railroads.

Co-operative action with the A. A. R., referred to in the previous report, has been continued for the purpose of improving the condition of couplers, draft gears, and their attachments and supports, with a view to eliminating or reducing the parting of trains due to slip-over of knuckles "which at prevailing high speeds with long trains is a serious hazard in operation." Note is taken of the fact that the deadline date for the elimination from interchange service of cars equipped with arch-bar trucks was extended by the A. A. R. to last December 31 with a definite provision that there would be no further extension. Also mention is made of the Bureau's co-operation with A. A. R. in developing revised specifications for the structural design of new passenger-train cars and "the importance of providing braking methods and apparatus to control high-speed trains properly."

"Considerable activity," the report goes on, "is evident in the development of new brake designs of disk, rotor and drum types which do not utilize treads of car wheels as braking surfaces, as well as further improvements in valvular mechanisms, devices to prevent sliding of wheels and means for sanding rails under all wheels simultaneously with brake applications, in order to provide the basic essentials of adequate retardation rate and rail-wheel adhesion. As yet these experiments and tests have not proceeded far enough to warrant any conclusions as to the relative efficiency of proposed systems of this type as compared with brake systems now in service." The Bureau's observations of these tests will continue as will its scrutiny of the results of last year's A. A. R. tests of trucks designed for high-speed freight service.

The section of the report on the Bureau's accident-investigation work reveals that during the 1938-39 fiscal year the commission received reports of 1,298 collisions and 3,121 derailments; in these 133 persons were killed and 828 injured, as compared with 195 killed and 1,115 injured in 1937-38's 1,469 collisions and 3,823 derailments.

Equipment Purchasing and Modernization Programs

Baltimore & Ohio.—The Baltimore & Ohio has asked the Interstate Commerce Commission to approve a plan whereby it would issue and sell to the Reconstruction Finance Corporation $5,330,000 of 2½ per cent equipment trust certificates, maturing in 10 equal annual installments of $533,000 on February 1, in each of the years from 1941 to 1950, inclusive. The proceeds would be used in part payment of the purchase price of 500 steel box cars, 500 gondola cars, 1000 steel hopper cars, and 100 steel hopper cars for cement loading, costing a total of $5,929,000. Orders for most of this equipment were reported in the November, 1939, issue of the *Railway Mechanical Engineer.*

Missouri Pacific.—The 1940 budget for the Missouri Pacific, calling for expenditures of $10,888,000 for improvements and betterments has been approved by the federal district court. Of this sum, $1,431,490 was authorized last October for the purchase of new rails and fastenings. Among the items in the budget are $1,350,000 for the purchase of new rolling stock; $250,000 for the purchase of a three-unit streamline train for operation between Memphis, Tenn., and Lake Village, Ark.; $580,000 for the purchase of 11 Diesel-electric switching locomotives; $570,000 for the purchase of 200 stock cars for the Texas lines; $390,580 for the rebuilding of four passenger locomotives and two freight locomotives; and several hundred thousand dollars for general repairs to 1,975 freight cars, and the installation of new type brakes on 1,210 freight cars. Sums are also included in the budget for the laying of new rail, station improvements, traffic control, and signal work.

The New York Central.—The New York Central contemplates buying a number of new passenger cars and plans to rebuild a number of its present cars.

Northern Pacific.—The Northern Pacific will purchase seven Diesel-electric switching locomotives. Three 1,000-hp. locomotives will be assigned to Northtown yards, Minneapolis, Minn., two 660-hp. to Seattle, Wash., and one each of 660-hp. to Spokane, Wash., and Tacoma. These locomotives are a part of the road's 1940 rail and equipment purchase program which will total $8,500,000.

St. Louis-San Francisco.—The 1940 budget of the St. Louis-San Francisco provides for the expenditure of $3,444,010 for mechanical and roadway improvements. The mechanical improvements include the converting and repairing of locomotives, $746,204; the rebuilding and remodeling of freight cars, $419,499; the rebuilding and remodeling of passenger cars, $198,778; and new shop machinery and equipment, $22,441. Total mechanical expenses for the year will be $1,386,922, not including $51,163 for Texas lines.

Southern Railway.—This company is undertaking to modernize in its own shops, 14 all-steel passenger coaches, which will be assigned to service between Washington,

Orders and Inquiries for New Equipment Placed Since the Closing of the January Issue

LOCOMOTIVE ORDERS

Road	No. of Locos.	Type of Locomotive	Builder
Chilean State Railways	10	4-8-2	American Locomotive Company

LOCOMOTIVE INQUIRIES

Road	No. of Locos.	Type of Locomotive	Builder
Terminal R. R. of St. Louis	5	0-8-0	....

FREIGHT CAR ORDERS

Road	No. of Cars	Type of Car	Builder
General Chemical Co.	75	Tank	Gen. American Trans. Corp.
Lehigh Valley	24	Caboose	Company Shops
Minneapolis & St. Louis	10	Hopper	Gen. American Trans. Corp.
Norfolk & Western	100*	Box	Greenville Steel Car Co.

PASSENGER CAR ORDERS

Road	No. of Cars	Type of Car	Builder
C. B. & Q.	5†	....	Edw. G. Budd Mfg. Co.

* One-half to be equipped with auto loaders.

† Tenth Zephyr.—A five-car Diesel-electric streamline train to be placed in operation next spring on a daily round trip schedule between Lincoln, Neb., and Kansas City, Mo., via Omaha, Neb. It will supplant the "Pioneer Zephyr," which is to be assigned elsewhere. The new train will consist of a 2,000-hp. Diesel-electric locomotive to be built by the Electro-Motive Corp., a combination mail and baggage car having a 30 ft. railway post office, a baggage and express car, two deluxe chair cars of 52-passenger capacity, and a diner-observation car with 24 dining seats and 22 parlor car chairs. Its name will be determined by a contest. Zephyrs eleven and twelve will be ordered later for delivery in mid-year. These eight-car trains will be known as the Texas Zephyrs and will operate between Denver, Colo., and Ft. Worth, Tex. and Dallas. Each train will include a 4,000-hp. Diesel-electric locomotive, a mail-express car, a baggage-coach, two deluxe chair cars, a dining-lounge car, all of stainless steel, and three standard Pullmans.

D. C., and Birmingham, Ala., and between Washington and Atlanta, Ga. The work, which is estimated to cost approximately $275,000, will include air conditioning, reclining chairs and complete new interior fixtures and decoration throughout.

Fusion-Welded Tank Cars

THE American Car & Foundry Co. and E. I. duPont deNemours & Company, Inc., have been authorized by the Interstate Commerce Commission to construct 36 fusion-welded tank cars for experimental service in the transportation of various corrosive materials.

The General American Transportation Corporation and the American Car & Foundry Co. have been authorized by the Commission to construct 75 fusion-welded tank cars for experimental service in the transportation of caustic soda solution and petroleum products.

Diesel-Electric Passenger Locomotive Built by Alco

THE first Diesel-electric passenger locomotive to be built by the American Locomotive Company has been delivered to the Chicago, Rock Island & Pacific, where it is being used on the Rockets. This locomotive is equipped with two 1,000-hp. MacIntosh & Seymour six-cylinder, vertical, four-cycle engines.

Supply Trade Notes

C. H. McCOLLAM, metallurgist of the steel and tube division of the Timken Roller Bearing Company, has been appointed assistant director of steel sales, with headquarters at Canton, Ohio.

◆

IRON & STEEL PRODUCTS, INC., Chicago, has established a merchant iron and steel department under the direction of J. C. Beggs, formerly of Joseph T. Ryerson & Sons, Inc.

◆

R. A. CANNON, vice-president in charge of casting sales of the Birdsboro Steel Foundry & Machine Co., Birdsboro, Pa., has been appointed vice-president in charge of sales. Mr. Cannon, who is now responsible for the entire sales activities of the company, entered the service of the Birdsboro Steel Foundry & Machine Company in 1921 and became vice-president in charge of casting sales in 1929.

◆

DR. A. LLOYD TAYLOR, formerly director of the Department of Chemistry, Pease Laboratories, New York, has joined the technical staff of the Oakite Products, Inc., New York. Mr. Taylor will concentrate primarily on chemical research and development of cleaning materials for production and related cleaning operations of major industries.

◆

HOWARD R. HAFFERKAMP has been appointed supervisor of purchases on the staff of the Bendix-Westinghouse Automotive Air Brake Company, Pittsburgh, Pa. Mr. Hafferkamp will have his headquarters in the company's manufacturing division at Wilmerding, Pa.

◆

W. B. MOORE, who has been associated for 20 years with The Timken Roller Bearing Company, Canton, Ohio, in various sales activities, has been appointed director of sales of the Steel and Tube division. Mr. Moore joined The Timken Company organization as an engineer early in 1919, following his graduation from the University of Michigan. After serving in the Canton office, he was placed in charge of the company's Pacific Coast district and since 1933 has been manager of industrial sales. S. C. Partridge, assistant manager of industrial sales, succeeds Mr. Moore as manager of industrial sales.

SIDNEY D. WILLIAMS, director of sales for the Timken Steel & Tube Division of the Timken Roller Bearing Company, Canton, Ohio, has been appointed vice-president in charge of sales for the new steel division at Warren, Ohio, of the Copperweld Steel Company, Glassport, Pa. Mr. Williams was graduated from Lehigh University in 1913 with the degree of metallurgical engineer. He worked in the various departments of the Carnegie Steel Company, Homestead, Pa., until 1918, during which year he served in the United States Naval Flying Corps. He then served, successively, as superintendent of the openhearth department of the Central Iron & Steel Co., Harrisburg, Pa., and as superintendent of the openhearth department and chief metallurgist of the Pittsburgh Crucible Steel Company, Midland, Pa. From 1926 to 1940 he was, in turn, metallurgical sales engineer, assistant director of sales, manager of tube sales, and director of sales for the Timken Steel and Tube division of the Timken Roller Bearing Company, Canton, Ohio.

Photo by Parry

S. D. Williams

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R. C. NORBERG, vice-president and general manager of the Electric Storage Battery Company, Philadelphia, Pa., was elected president and general manager at a recent meeting of the board of directors. John R. Williams, who has been associated with the company for the past 45 years, has retired as president.

HOWARD A. FLOGAUS, assistant to vice-president of the J. G. Brill Company, Philadelphia, Pa., has been appointed chief engineer, with headquarters at Philadelphia.

◆

J. HOMER PLATTEN has been elected a vice president of the American Car and Foundry Company, New York. Mr. Platten will continue as comptroller of the company but will relinquish the office of executive assistant to the president.

◆

R. L. SALTER has been appointed engineer of tests in charge of specifications and inspections of the Association of Manufacturers of Chilled Car Wheels. The inspection department of the association is represented by resident inspectors stationed at member companies' foundries, and the finished product is subject to the association's inspection and test before it can be released for shipment.

◆

W. W. WILLIAMS, general manager of The Babcock & Wilcox Tube Co., Beaver Falls, Pa., will relinquish his position on March 1, to go into his own business on the Pacific Coast. Mr. Williams became associated with The Babcock & Wilcox organization in 1929, as sales counsellor and in turn became general sales manager and general manager of the Babcock & Wilcox Tube Co.

◆

THE NATIONAL BRAKE COMPANY, INC., will move its executive office from Buffalo, N. Y., to 50 Church street, New York, effective March 1. William M. Wampler has been elected president of the company with C. T. Stansfield, vice-president; E. C. Mersereau, vice-president in charge of sales and S. T. Pearson, secretary and treasurer.

◆

JOHN W. LOHNES, assistant to general manager of sales of the Vanadium Corporation of America, has been appointed assistant general manager of sales, with headquarters at New York. Donald C. Hostettler, eastern sales representative, has been transferred to Detroit, Mich., to take charge of the Detroit district sales office as sales representative to succeed J. Berens Waters, who is assuming the duties of general purchasing agent. John B. Girdler has been appointed eastern sales representative at New York to succeed Mr. Hostettler.

(Turn to second left-hand page)

NEREUS HUBERT ROY has recently been appointed head of the Research Department of the Waugh Equipment Company, New York. Mr. Roy was born in Bonham, Tex., on July 4, 1895. He is an authority on instruments for, methods of, and practical interpretation of investigations in connection with mechanical equipment, such as passenger and freight cars, locomotives, draft gears, rails, wheels, axles and trucks. He attended the University of Texas from 1913 to 1917, and was commissioned and served in the U. S. Army Engineers, 1917-19. To pursue more intensive study of specific engineering problems he returned to college, receiving the degree of B. S. in C. E. from the University of Texas in 1920, and the degree of B. S. in C. E. (reciprocal) from the University of Mexico in 1923. He carried on subsequent post graduate work at the University of Texas and the University of Illinois, obtaining from the latter the degree of M. S. in Theoretical and Applied Mechanics in 1929 and the Professional Degree of C. E. in 1930. During this period (1920-24) Mr. Roy had been engaged as field engineer and acting chief engineer of the International Petroleum Company, Tampico, Mex., and (1925-28) as chief engineer and general field superintendent of the Milham Corporation of Texas (Seaboard Oil Corporation). He joined the faculty of the University of Illinois, Engineering Experiment Station, in 1928 and was appointed assistant research professor in 1931, in which position he remained until the Fall of 1937. While at the University of Illinois he carried on research work on railroad engineering

N. H. Roy

problems, including the study of fissures in rails, stresses in car axles, stresses in passenger and freight cars under impact, and locomotive balance. He was engaged as research consultant by the Association of American Railroads, the Pullman Standard Car & Manufacturing Company, and several railroads. On January 1, 1938, he entered the engineering department of the Waugh Equipment Company and was appointed director of research on November 1, 1939. Mr. Roy is a member of the Western Railway Club, the New York Railroad Club, the American Association for the Advancement of Science, and Sigma Xi, National Honorary Research Society.

Obituary

AMBROSE N. DIEHL, who was president of the Columbia Steel Company from 1932 to September, 1939, died on January 3, at La Jolla, Cal., from bronchial pneumonia.

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CHARLES K. KNICKERBOCKER, first vice-president of the Griffin Wheel Company, Chicago, died on January 7 of pneumonia. He had been ailing for several years. Mr. Knickerbocker was born in Chicago on September 28, 1874, and entered the employ of the Griffin Wheel Company on November 1, 1894, as a shipping clerk. He was promoted to sales agent in 1895, to general sales agent in 1909, and to first vice-president in 1914. He had also been a director since 1914.

Personal Mention

General

T. R. FREDERIKS, assistant dynamometer engineer of the New York Central, has been appointed dynamometer engineer, with headquarters in New York, succeeding J. J. Anderson.

J. J. ANDERSON, dynamometer engineer of the New York Central, has been appointed assistant to assistant chief engineer motive power and rolling stock, with headquarters in New York, succeeding W. C. Wardwell.

W. C. WARDWELL, assistant to assistant chief engineer of motive power and rolling stock of the New York Central, with headquarters at New York, has been appointed assistant engineer of tests with the same headquarters, succeeding W. F. Collins.

H. T. COVER, master mechanic of the Columbus, Cincinnati and Toledo divisions of the Pennsylvania, at Columbus, Ohio, has been appointed superintendent of the Wilkes-Barre division, with headquarters at Sunbury, Pa., succeeding R. W. Sheffer. Mr. Cover was born in Altoona, Pa., on August 25, 1897, and entered the service of the Pennsylvania as a laborer on the Middle division at Altoona in August, 1915. After two weeks' work as a laborer, he was transferred to the Juniata shops in Altoona as a boilermaker helper. On June 12, 1917, he became a draftsman in the office of the general superintendent of motive power, and on January 16, 1920, was appointed a special apprentice in the Altoona machine shop. On October 18, 1922, he became a motive-power inspector, assigned to the office of the chief of motive power in Philadelphia. On May 1, 1923, Mr. Cover became assistant shop foreman on the New York division, and on July 16, 1924, advanced to the position of foreman on the same division. On May 1, 1927, he was

H. T. Cover

appointed shop foreman on the Philadelphia Terminal division, and on June 16, 1929, returned to the East Altoona enginehouse. On January 1, 1931, he was promoted to assistant master mechanic of the Maryland division at Wilmington shops, and on November 1, 1934, became master mechanic on the Buffalo division. On April 16, 1937, he was assigned to the Maryland division, and on July 1, 1939, to the Columbus, Cincinnati and Toledo divisions, with headquarters at Columbus.

DANIEL K. CHASE, superintendent of the Pittsburgh division of the Pennsylvania, has been appointed general superintendent of the eastern Ohio division, with headquarters as previously at Pittsburgh, Pa. Mr. Chase was born on June 1, 1896, at Rehoboth, Del. He attended the public schools of Rehoboth Beach and Lewes, Del., and was graduated from Pennsylvania State College (B. S. M. E., 1922). He began his railroad career on December 16, 1913, as a messenger in the transportation department of the Pennsylvania at Philadelphia, Pa. In July, 1914, he was transferred to the Altoona Works as a machinist apprentice, subsequently being appointed special apprentice, completing his apprenticeship on April 16, 1922. During this time Mr. Chase attended college, continuing his employment during vacations. After service in the World War he resumed his employment with the Pennsylvania and continued his course at Pennsylvania State College. Following service in several division and regional offices as motive-power inspector, he was appointed assistant master mechanic at Jersey City, N. J., on November 1, 1923, later serving in a similar capacity at Wilmington, Del., and Altoona, Pa. On March 1, 1927, Mr. Chase was appointed master mechanic, serving successively at Olean, N. Y.; Can-

ton, Ohio; Jersey City, N. J.; Chicago and Pittsburgh. On November 1, 1934, he was appointed superintendent, Toledo division, Toledo, Ohio; on November 1, 1935, superintendent, Eastern division, with headquarters at Pittsburgh, Pa., and on September 16, 1939, superintendent of the Pittsburgh division.

Master Mechanics and Road Foremen

H. D. ALLEN, road foreman of engines of the Ft. Wayne division of the Pennsylvania, has been appointed road foreman of engines of the Cleveland division.

J. E. MCLEOD, assistant master mechanic of the Chesapeake & Ohio at Peru, Ind., has been appointed master mechanic of the Chicago division, with headquarters at Peru.

W. R. DAVIS, master mechanic of the Chicago Terminal division of the Pennsylvania, has been transferred to Harrisburg, Pa., replacing W. B. Porter, who has been transferred.

F. R. KIRKPATRICK, assistant road foreman of engines of the Pittsburgh division of the Pennsylvania, has been appointed road foreman of engines of the Renova division.

P. T. BRIERS, general foreman of the Chesapeake & Ohio, at Hinton, W. Va., has been appointed master mechanic of the Cincinnati division, with headquarters at Covington, Ky.

J. W. LEONARD, assistant engineer of motive power of the Pennsylvania at the Altoona Works (Altoona, Pa.), has been appointed master mechanic at Chicago, succeeding W. R. Davis.

J. D. COUSINS, road foreman of engines of the Cleveland division of the Pennsylvania, has been appointed road foreman of engines of the Indianapolis division, with headquarters at Indianapolis, Ind.

J. A. WARREN, road foreman of engines of the Indianapolis division of the Pennsylvania, has been appointed road foreman of engines of the Ft. Wayne division, with headquarters at Ft. Wayne, Ind.

FRANCIS H. WINGET, who has been appointed master mechanic of the New York Central at Bellefontaine, Ohio, as announced in the January issue of the *Railway Mechanical Engineer*, was born at Pennville, Ind., on October 31, 1901. He attended grade and high schools at Pennville and was graduated in mechanical engineering from Purdue University in June, 1923. Mr. Winget entered railroad service in the employ of the Cleveland, Cincinnati, Chicago, & St. Louis on June 18, 1923, as a special apprentice at the Shelby street enginehouse, Indianapolis, Ind., subsequently serving in the shops, enginehouse and test departments at Mt. Carmel, Ill., and Beech Grove, Ind. From December, 1926, to October, 1932, he was an inspector on special assignments for the master mechanic at Bellefontaine. In October, 1932, Mr. Winget was transferred to Indianapolis as a special inspector, working directly for the superintendent of equipment. From May, 1935, until October, 1937, he was lubrication inspector, working out of Indianapolis. In October, 1937, he became assistant enginehouse foreman at Shelby street enginehouse and on April 1, 1938, was appointed assistant master mechanic of the Ohio division, with headquarters at Bellefontaine. He became master mechanic on October 1, 1939.

W. B. PORTER, master mechanic of the Philadelphia division of the Pennsylvania at Harrisburg, Pa., has been transferred to the Columbus, Cincinnati and Toledo divisions, with headquarters at Columbus, Ohio, succeeding H. T. Cover.

J. E. GARRETSON, assistant master mechanic of the Chesapeake & Ohio at Russell, Ky., has been appointed master mechanic of the Russell and Ashland divisions, with headquarters at Russell, succeeding R. G. McKee, deceased.

F. J. TOPPING, assistant master mechanic of the Chesapeake & Ohio at Stevens, Ky., has been appointed master mechanic of the Hinton division, with headquarters at Hinton, W. Va., succeeding W. V. Hinerman, whose promotion to assistant to the superintendent of motive power, was announced in the January issue.

J. ALLEN CLARK, who has been appointed master mechanic of the Northern Pacific as announced in the January issue of the *Railway Mechanical Engineer*, began his career with the Northern Pacific in 1902 on the Tacoma division. He was road foreman on the Tacoma division during 1930 and most of 1931. He returned to the cab of a locomotive during the summer of 1931, and in March, 1935, was transferred to the position of road foreman on the Rocky Mountain division, with headquarters at Livingston, Mont. He was transferred to Missoula, Mont., in March, 1938, and on November 1, 1939, became master mechanic of the Idaho division, with headquarters at Parkwater, Wash.

Car Department

WALTER E. DUNHAM, superintendent of the car department of the Chicago & North Western and the Chicago, St. Paul, Minneapolis & Omaha at Chicago, has retired.

Shop and Enginehouse

E. L. LAM, gang leader in the electrical department of the Norfolk & Western at Roanoke, Va., has been promoted to the position of assistant foreman in the erecting shop.

Obituary

CHARLES F. GILES, who retired in August, 1928, as superintendent of machinery of the Louisville & Nashville, with headquarters at Louisville, Ky., died at his home in Louisville on January 13. He had been virtually an invalid for a number of years. Mr. Giles was born at Rowlesburg, W. Va., on November 2, 1856, and entered railway service in 1873 as a machinist apprentice on the Baltimore & Ohio at Wheeling, W. Va. From 1877 to 1882, he was a machinist on this road, the Texas & Pacific, the Pennsylvania and the L. & N. and from the latter date until 1887, he was an enginehouse foreman and machine shop foreman on the latter road. In 1887, he was promoted to master mechanic at Birmingham, Ala., and on October 1, 1902, he was transferred to Louisville. On February 1, 1904, he became assistant superintendent of machinery and on June 30, 1911, he was promoted to superintendent of machinery.

ROBERT CHARLES GOREY, SR., who retired as assistant master mechanic of the Louisville & Nashville at Montgomery, Ky., on May 1, 1916, died on November 28, at the age of 82.

MAX H. WICKHORST, former engineer of tests of the Chicago, Burlington & Quincy, with headquarters at Aurora, Ill., and from February, 1910, to June, 1922, engineer of tests for the Rail committee of the American Railway Engineering Association, with headquarters at Chicago, died at Oak Park, Ill., on January 2.

CHARLES J. SCUDDER, consulting engineer of motive power of the Delaware, Lackawanna & Western, died at his home in Scranton, Pa., on Tuesday, January 30. Mr. Scudder was born at Saginaw, Mich., on September 21, 1873, and entered railway service in 1888 as a machinist apprentice on the Flint & Pere Marquette (Pere Marquette). In 1898, he became a machinist on the Detroit, Grand Rapids & Western (Pere Marquette) at Ionia, Mich., and in 1906, became master mechanic on the

Charles J. Scudder

Cincinnati, Hamilton & Dayton (Baltimore & Ohio) at Cincinnati, Ohio. Mr. Scudder was appointed general foreman, Pere Marquette, at Chicago, in 1908; superintendent shops at Saginaw, Mich., in 1909; and master mechanic in 1910. In 1911 he became a locomotive inspector of the Interstate Commerce Commission and in 1917 was appointed supervisor of equipment, United States Railroad Administration. Mr. Scudder was appointed superintendent of shops for the Delaware, Lackawanna & Western at Scranton, Pa., in 1919 and became superintendent motive power and equipment at Scranton in 1923. Last fall he retired as chief of motive power, to which position he was appointed in 1936.